THE ENGINEERS' DIGEST

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EDITORIAL

IT is a well-known fact that the volume of available technical literature is steadily increasing and, according to the latest information, 18,000 trade and technical publications are regularly issued all over the world. This represents an increase of about 30 per cent over the pre-war volume. Therefore, it is becoming increasingly difficult for the progressive engineer to keep pace with the latest developments of immediate concern to his activities. This may well be a contributory factor in the present-day tendency towards specialisation. We realise that the complexity of various engineering subjects demands a persistent and devoted study in order to master any particular field. The result is that industry today is dependent on a great number of specialist engineers who, although highly competent in their own respective branches, have lost sight of important developments in other fields, which, however, may frequently have a definite bearing on their work.

There are, today, few engineering products involving problems which can be tackled by one man alone. Mechanical and electrical engineers, metallurgists, chemists, physicists, and electronic engineers, as well as industrial designers, may all contribute in some degree to the creation of the final product. It might well be asked, then, why this journal deprecates the trend towards specialisation indicated. The answer is, of course, that while realising the necessity for specialisation, we cannot agree that efficient teamwork is possible unless every member of the team is familiar at least with the basic possibilities of his colleagues' work and with its attendant problems. This explains our policy to stimulate reader interest in a wide variety of subjects and to keep the reader informed of the latest technical advancements in all fields of engineering and in all parts of the world.

While it should be the duty of every engineer to devote a fair part of his time to reading the technical press, the engineering executive finds it extremely difficult to allocate even a strictly limited amount of time to this task, mainly because of the wide scope of his responsibilities. This does not mean that the industrial executive has lost any interest in the broad aspects of technical progress. Large piles of periodicals accumulating on his desk furnish ample evidence of his intentions, but also of his lack of reading time. We are able to sympathize with his heavy burden and, up to now, could well understand his attitude.

In this connection, it is worth remembering that, according to authoritative estimates, 99 per cent of the information published either deals with known developments or is of insufficient importance to merit careful study. Whilst we have no intention of minimising the usefulness of those publications which mainly offer detailed descriptions of existing installations and engineering products already on the market, we contend that the need exists for a technical journal which makes an attempt to keep its editorial within the limits of the one per cent of published information of vital importance to well-informed and progressive technical men. This policy has been pursued by " The Engineers' Digest " since its inception, and is being increasingly appreciated by engineers all over the world.

There are, however, industrial executives who cannot even spare the time to acquire an overall appreciation of the general trends of engineering science from the articles contained in our main editorial columns. For these executives we now offer, as a remedy to this situation, a concise, yet comprehensive, survey of the latest technical developments of international significance. They will be critically selected from a wealth of material and presented each month as brief items of not more than 200 words each, in simple, non-technical language, explaining their relative merits and possible applications.

Our survey is introduced in this issue in a special section called THE BLUE PAGES. A special organisation within the framework of " The Engineers' Digest " has been set up to collect and sift information from thousands of different sources. The result of this research will be found in THE BLUE PAGES, which, we hope, will prove of great benefit to all those who can spare only a few minutes each month for reading technical literature, yet wish to be well informed.

GERMANY

A New Wall-Thickness Gauge for Ferromagnetic Materials

By R. BERTHOLD. (From *Stahl und Eisen*, Vol. 70, No. 6, March 16, 1950, pp. 233-234, 4 illustrations.)

IN order to determine the wall thickness of metal bodies which are only accessible on one side, various methods have been developed, the most notable being: (1) the ultrasonic reflection method, working on the echo sounding principle; (2) methods using a Geiger-type counter, which indicates the radiation transmitted by the test piece when the latter is subjected to the radiation produced by a radio-active preparation; (3) potentiometric methods, in which the electrical conductivity serves as a measure of the wall thickness; and (4) magnetic induction methods, in which the permeance of the material is used to indicate wall thickness. All these methods have their advantages and limitations, and the great variety of materials, shapes and surface finishes of structural components precludes the possibility of one universal method being suitable for all purposes.

A new and particularly simple method, which, however, can only be used for ferromagnetic materials of moderate thickness, will now be described. With this method, the wall thickness is determined by measuring the magnetic field energy which is liberated when switching off a d.c. electromagnet placed on the specimen being tested. The electromagnet is set in position so that it will be in close contact with the surface of the test specimen, and the energising current is then switched on. The magnetic field thus generated depends, among other things, on the magnetic path within the specimen, closing the magnetic circuit, and hence on the thickness of the specimen. On switching off, the magnetic field breaks down and only the residual field remains; this residual field generates a current pulse in a measuring coil mounted on one of the pole pieces of the magnet. A compensating coil, mounted over the yoke of the magnet and acting in opposition to the measuring coil, is also used in order to eliminate the effects of stray fields not passing through the cross-section of the specimen, and the compensating coil is adjusted so that zero deflection is obtained for zero wall thickness.

Experimental results have shown that this equipment has a linear response as long as the magnetised test section of the specimen is fully saturated. When this condition is not fulfilled, the sensitivity decreases rapidly and the instrument reading is then no longer proportional to the thickness of the material. The linear measuring range depends, therefore, on the core cross-section and the number of ampere-turns of the electromagnet, and is, therefore, limited by the weight of the measuring magnet. The number of ampere-turns should thus be well above the normally required value in order that fluctuations in the coil voltage may have no effect on the reading. In the linear range the measuring accuracy is remarkably good, and reaches approximately $\pm$ 2 per cent.

Typical results obtained with this equipment are given in the following table:—

Test specimens	Cross-sectional area of magnet core (mm^2)	Maximum wall thickness for linear response (mm)
flat sheets* ..	950	4
flat sheets* ..	2300	8
tubes	2300	5

*Rolled sheets of unalloyed structural steel with a U.T.S. exceeding 34 kg/mm^2.

Disturbing influences which may affect the readings are primarily due to variations in the air gap, which can vary with the position of the pole pieces on the specimen when the test surface is scaly or rusty or shows signs of corrosion. The variations in sensitivity due to changes in the air gap, which can be overcome to a certain extent, are shown in the following table:—

Air gap (mm)	Instrument readings (arbitrary divisions)	
	Core cross-section	
	950 mm^2	2300 mm^2
0	0	0
0·1	19·0	
0·2	20·0	20·5
0·3	20·2	
0·4	20·0	22·7
1·0	16·5	25·2
1·4		25·0
1·8		24·4
2·2		23·3
2·6		21·75
3·0		20·0

It may seem surprising that the instrument reading and, therefore, the magnetic energy liberated while switching off increase at first with increasing values of the air gap, as the total energy of the magnetic field must be lower. However, this effect takes place because the residual field at first decreases more rapidly than the total magnetic field, so that the result is an increase in the liberated magnetic field energy. When the air gap reaches greater values, the decrease of the total energy becomes preponderant, and the sensitivity curve thus has a region with a fairly flat maximum, which is the most suitable region for obtaining good measuring accuracy.

Thus, it is only necessary to arrange for an air gap of a suitable value in front of the pole pieces (for instance, by using non-ferromagnetic supporting surfaces), in order to eliminate within fairly wide limits the effect of air-gap variations. For wall thicknesses below 3 mm, the standard pole pieces can then be applied directly on either flat or tubular specimens. For greater wall thicknesses, it is necessary to fit the pole pieces with adaptor shoes having the same curvature as the surface of the specimen.

The calibration curve will be different according to the permeability and curvature of the material tested. The most convenient arrangement is to place the magnet on the specimen so that it will be parallel to the longitudinal axis of the tube. Rust deposits on the internal surface of the tube or container will have no practical

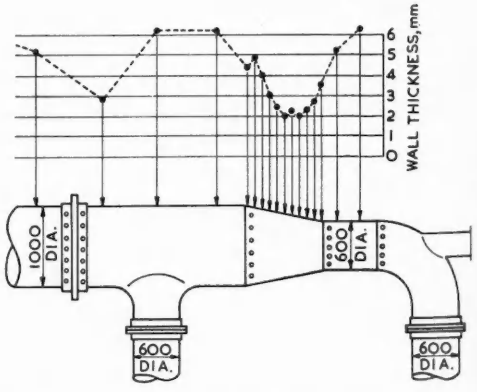


Fig. 1. **Wall thickness measurements on a water pipe system**

effect on the measurement results, as the permeability of loose iron oxide layers is much lower than that of the iron cross-section of the specimen.

An example of the results obtainable by means of this non-destructive test method is illustrated in Fig. 1, which shows the wall thickness measurements obtained on a pipe system supplying cooling water to two turbines. The measurements were taken while the turbine plant was in operation. The nominal wall thickness was 6 mm for all ducts. At each station along the pipe sections being investigated, the magnetic gauge was placed at three different points of the periphery but situated in the same vertical plane, and average values were taken for each station. The results given in Fig. 1 show considerable deviations from the nominal thickness value, both in the main duct and along the tapered portion, where in some places the wall was only 2 mm thick. The variations of the wall thickness and the minimum thickness positions clearly indicated corrosion effects due to eddy formations in duct sections in which there was a change in the velocity or direction of the water flow.

SWITZERLAND

The Neutralisation Value as a Measure of the Stability and Miscibility of Mineral Oils

By L. STANISAVLIEVICI. (From *Schweizer Archiv*, Vol. 16, No. 1, January, 1950, pp. 9-21, 31 illustrations).

THE artificial ageing method developed by Weiss and Salomon of the Ecole de Pétrole, Strasbourg, has been used successfully in many countries for predicting the stability of insulating and steam turbine oils which have been in service over a number of years. In this method of testing, slightly modified by the author, the oil is maintained at 115° C in the presence of air and a copper catalyst, and petroleum ether insolubles are determined at intervals.

The neutralisation value gives reliable information on the progress of deterioration of insulating and steam turbine oils in actual use. It is more quickly and easily determined than petroleum ether insolubles and can be repeated, if required. This fact induced the author to examine the possibility of using the neutralisation value in conjunction with the " Strasbourg " stability test.

First, a number of Rumanian insulating and steam turbine oils was artifically aged by the Strasbourg method. Neutralisation values of the insoluble matter formed were determined and found to be practically constant, although the amount of insoluble matter increased considerably with time. This confirmed the probability that the neutralisation value may be suitable for measuring the progress of deterioration in the Strasbourg test.

Further artificial ageing tests were made with a number of Rumanian steam turbine and insulating oils. The proportion of insoluble matter and the neutralisation values were plotted against duration of test. In each case, the neutralisation value varied in a manner very similar to that of the insoluble matter, as is shown, for example, in Fig. 1 for an oil of high quality and stability, and in Fig. 2 for an oil of poor quality and low stability.

It is concluded that the neutralisation value curves, together with the analysis data of the new oil, are a satisfactory indication of oil stability.

In the next stage of the work, seven modifications of the Strasbourg stability test were examined, such as the use of different sizes of beakers and heavier or differently shaped copper catalysts, and were compared with the original method. Four Rumanian insulating oils and one American and four Rumanian steam turbine oils were used. It was found that the conditions of test have a considerable effect on results, and that the formation of insoluble solids is particularly susceptible to variations of procedure, much more so than the neutralisation value. It is concluded that also from the point of view of accuracy of results the determination of the neutralisation value is superior to the determination of petroleum ether insolubles. Another conclusion reached is that the Strasbourg method may be simplified by using only one beaker for each oil under test, from which 5 cm³ samples are taken for determining neutralisation values as required.

Further work was concerned with the question of miscibility of insulating and steam turbine oils. This question frequently arises in service when an old filling needs topping up. Sixteen insulating oils of varying origin, new and used, all of commercial quality, were available. From these, 18 blends were prepared, the majority by mixing two oils of different quality. Another series consisted of blends of used oils, mixed in proportions met with in practice. Each blend and its components were tested by the Strasbourg method for

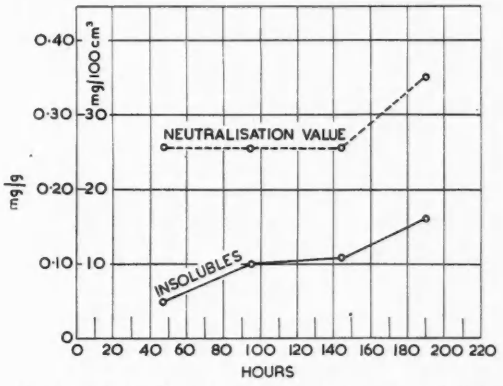


Fig. 1. **Deterioration of an oil of good quality and high stability.**

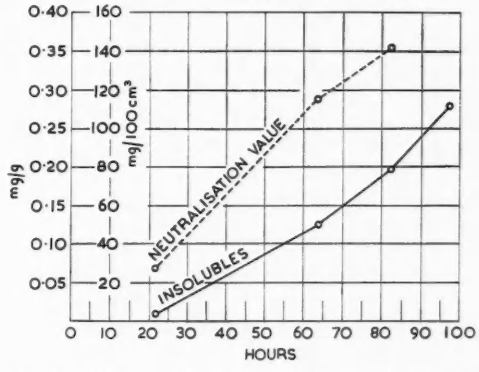


Fig. 2. **Deterioration of an oil of poor quality and low stability.**

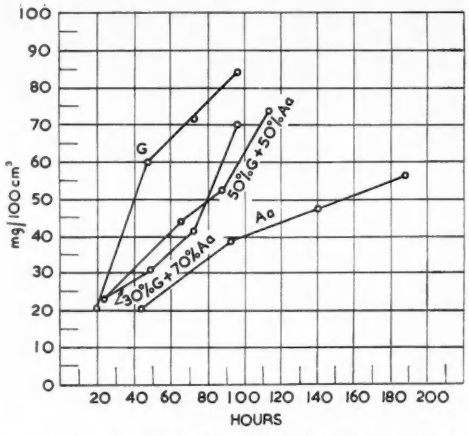


Fig. 3. **Petroleum ether insolubles of oils " G " and " Aa " and their blends.**

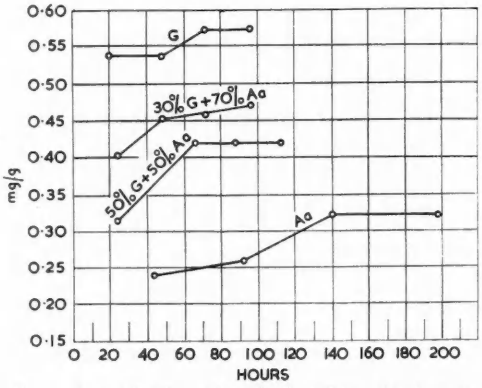


Fig. 4. **Neutralisation value of oils " G " and " Aa " and their blends.**

a period equivalent to at least four times the interval before the first appearance of cloudiness.

Out of the 18 blends, 17 had a stability higher than that of the lower quality component, and one was even better than the better quality component. One blend behaved similarly to the poorer quality component. These results, based on curves of petroleum ether insolubles, have been confirmed by the practical performance of four blends of used oils over the course of two years. They contradict the generally accepted view that a blend of two oils of widely different quality will give a performance similar to the low quality component.

It is concluded that blends of insulating oils of the same or different quality may have a stability better than that of the poorer quality or even that of the better quality component. The method appears to be suitable for predicting the stability of a blend, or even for choosing a blend of a desired stability.

Further experiments were devoted to the suitability of the neutralisation value as a criterion of miscibility. Fourteen new and used insulating and steam turbine oils of Rumanian, American, Italian and unknown origin were used. Nine mixtures were prepared in varying proportions, each consisting of two oils. These blends were treated by the Strasbourg method, and neutralisation values and petroleum ether insolubles were compared with those of their components.

Figs. 3 and 4 show typical curves for two oils, " *G* " and " *Aa*," and for their 50/50 and 70/30 blends' They indicate that the stability of the blends lies between those of the two oils. Moreover, the 50/50 blend, containing more of the low quality oil " *Aa*," appears to be preferable to the 70/30 blend.

A blend of 80 per cent of oil " *M* " and 20 per cent of oil " *E* " was not very different from its components as far as petroleum ether insolubles are concerned (see Fig. 5). Its neutralisation value, however, shown in Fig. 6, increased beyond that of either of its components when heated for longer than 50 hours, indicating a somewhat reduced stability.

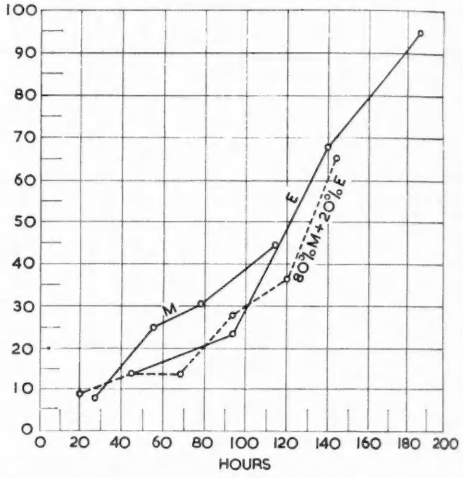


Fig. 5. **Petroleum ether insolubles of oils " E " and " M " and their blend.**

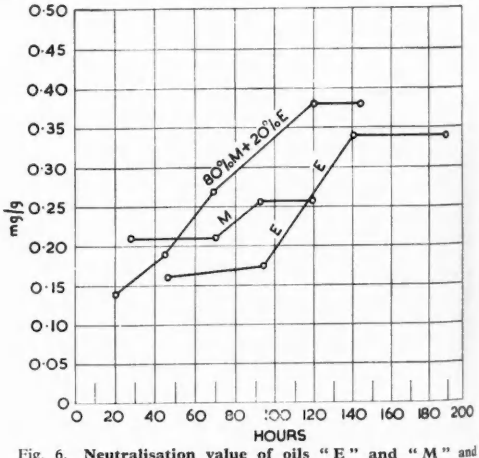


Fig. 6. **Neutralisation value of oils " E " and " M " and their blend.**

The agreement between neutralisation value and petroleum ether insolubles is satisfactory. Consequently, the former is suitable as an indication of the miscibility of mineral oils. In general, the curve of the blend should lie between those of its components. However, if the poorer quality component has sufficient stability, the curve of the blend may be as low as its own curve.

Electric Spark Machining

By I. Koncz. (From *Gép*, Vol. 1, No. 9-10, September-October, 1949, pp. 388-411, 38 illustrations.)

(*Concluded from April issue*).

EVALUATION OF THE OSCILLOGRAMS

Fig. 7 shows an oscillogram taken during a drilling process. It can be seen that during one period the condenser discharged four times across the imperfect and changing contact made between the electrode and component, and the charge q which passed this contact is

$$q = C(E_1 + E_2 + E_3 + E_4)$$

where $E_1 \ldots E_4$ are the peak values of the voltage previous to a discharge. The work N done in these discharges is

$$N = \frac{C}{2}\left[E_1^2 + E_2^2 + E_3^2 + E_4^2\right]$$

If the number of oscillograms taken during a drilling process was P and the time associated with a particular oscillogram is t_k, then during the time t_k, $t_k \times n$ was the number of contact cycles of the electrode holder; n is the number of contact cycles per sec. The charge Q_k passing the contacts in the time t_k is then:

$$Q_k = n t_k q_k$$

where q_k is evaluated from the respective oscillogram. The work done N_k is

$$N_k = n t_k N$$

The charge Q passing the contacts throughout the whole drilling process is

$$Q = n \sum_{k=1}^{P} t_k q_k$$

and the total work done N_e

$$N_e = n \sum_{k=1}^{P} t_k N$$

The average value of the potential e on the condenser is

$$e = \frac{2N_e}{Q}$$

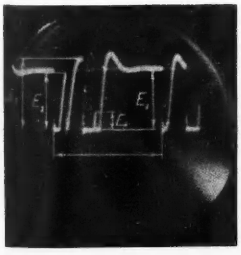


Fig. 7. **Evaluation of the voltage oscillogram.**

On the basis of these quantities further values were calculated, which are:

(1) The volume loss of electrode and component per unit charge;
(2) The volume loss of electrode and component per unit of work done;
(3) The volume losses in one period.

As an example, the evaluation of Fig. 7 gives these results:

$q_1 = 60\,(204 + 192 + 238 + 62)\,10^{-6} = 41{\cdot}76 \times 10^{-3}$ Coulomb

$N_{e1} = 4{\cdot}17$ Joule, $Q = 414{\cdot}7$ Coulomb, $N_e = 41400$ Joule.

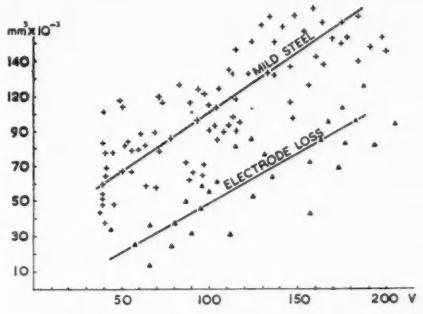


Fig. 8. **Volume loss per Coulomb.**

The material was Siemens-Martin steel, drilling time 6 minutes, the losses of the component 0·00131 lb, (0·00466 cu. in.), the losses of the electrode 0·0018 lb, (0·00367 cu. in.), the number of discharges 36,400, the average voltage $e = 200$ V.

The specific volume losses for the component are $3{\cdot}16 \times 10^{-6}$ lb/Coulomb, $3{\cdot}16 \times 10^{-8}$ lb/Wsec and $3{\cdot}6 \times 10^{-6}$ lb/discharge, and for the electrode $4{\cdot}34 \times 10^{-6}$ lb/Coulomb, $4{\cdot}34 \times 10^{-8}$ lb/Wsec and $4{\cdot}95 \times 10^{-8}$ lb/discharge.

The specific volume losses per unit charge of the component and the electrode are plotted in Fig. 8 against the voltage as abscissae. Individual measurements stray considerably. It appears that the specific volume losses per Coulomb are proportional to the voltage.

The meaning of the measurements can better be seen from Figs. 9 and 10. The specific volume losses per discharge are plotted against the voltage for different capacities for the component in Fig. 9 and for the electrode in Fig. 10. The curves are linearly proportional to the voltage at higher values, and their linear projections intersect on the abscissae at 55 Volts. This voltage can be observed in every contact period at the time of contact interruption. The specific loss also increases with the capacity.

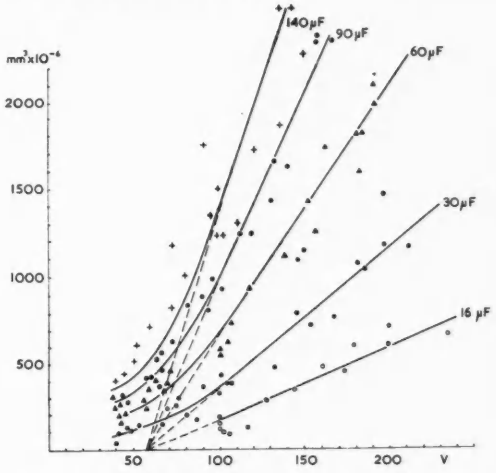


Fig. 9. **Volume loss of the component per discharge.**

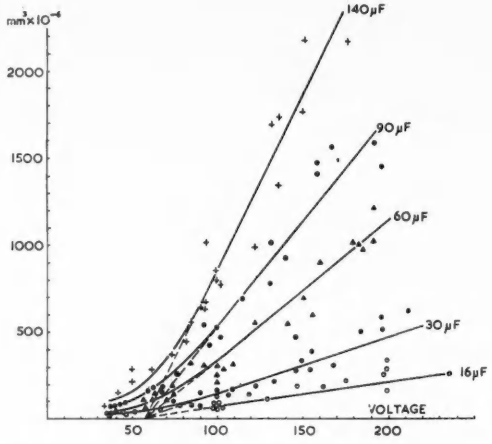


Fig. 10. **Volume loss of the electrode per discharge.**

The charge resistance had little influence on the output of the drilling process.

Fig. 11 shows the specific loss of the component per Wsec as a function of the voltage for different metals. This shows also that the output of this drilling process tends towards steady values with increasing voltage and is, therefore, independent of the capacity in this respect. The curves are similar for all metals, with the exception of that for cast iron, which is a straight line. The curves show that the efficiency of the process increases with decreasing voltage. On the other hand, the specific loss per discharge decreases heavily with decreasing voltage. This is shown in Figs. 12 and 13, where this loss per discharge is plotted against the voltage for different metals. Fig. 12 relates to the component and Fig. 13 to the electrode. At high voltages, the efficiency of the process is constant and the specific loss per discharge increases with the voltage.

The electrode loss is smaller when metals of lower tensile strength are drilled. The loss is higher when drilling metals which can be alloyed with tungsten, the metal of which the electrode is made. The electrode loss is higher when drilling cast iron than when drilling steel.

THE MECHANISM OF SPARK MACHINING

Figs. 14, 15, and 16 show oscillograms of the current. Two types of discharge phenomena can be observed. One is a damped harmonic oscillation of the current, and it is assumed that in this case the electrode sticks to the component and no machining is done. The cases where the component loses material are characterized by sharp current peaks, which die off at once. It is interesting to note that all current peaks during the drilling of a hole are equal in magnitude, independently of whether they remove material or not. Now it is known that the melting and current interruption of fuse wires, which in certain respects is a similar process to spark machining, take place at somewhat varying values on the ascending part of the current curve. As the maximum current values in spark machining are always equal, it is thought that it is not only the melting and the disintegration of the material which determine the time of interruption of this current.

Several of the oscillograms showed the formation of an arc when the contact was interrupted. The voltage peaks at that time are of the order of 50-75 volts. It is thought that the constant efficiency of the mechanism at higher voltages can be explained by the fact that an arc is always formed. This arc is not formed after every contact at lower voltages; hence the increasing efficiency. These arcs are, naturally, rapidly quenched. It has been observed in experiments, where the circuit data made the formation of an arc easily possible, that the electrode loss increased, but not so the component loss. Furthermore, in some cases the tungsten of the electrode became deposited on the walls of the hole. This is understandable if welding, the deposition of material, is considered to be the opposite process to spark-machining. Fig. 17 shows a comparison of welding and spark-machining, where the material addition and loss respectively are plotted against the input power and the much larger output of spark-machining is obvious. The direct current

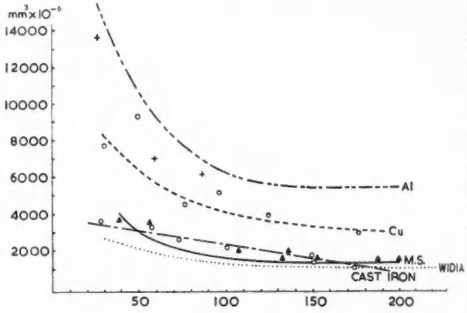


Fig. 11. **Volume loss of different component metals per Wsec.**

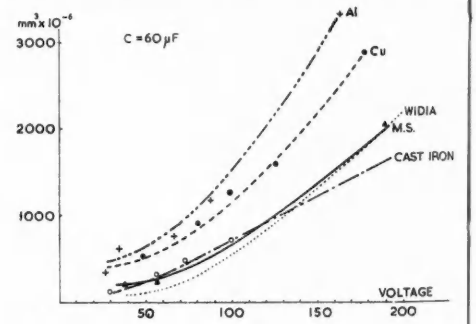


Fig. 12. **Volume loss of different component metals per discharge.**

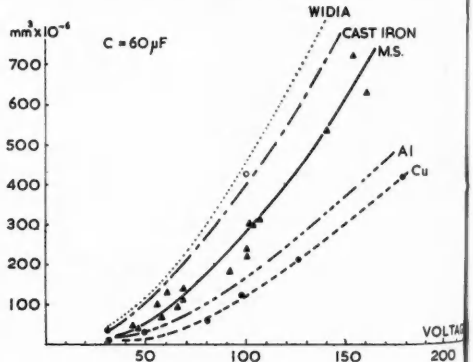


Fig. 13. **Volume loss of a tungsten electrode, drilling different component metals.**

Fig. 14

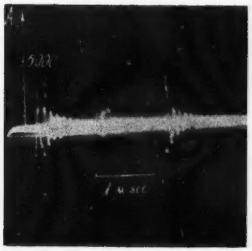


Fig. 15

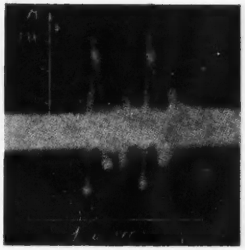


Fig. 16

Figs. 14, 15, and 16. **Current oscillograms.**

of arc formation is supplied direct from the generator and its voltage characteristic is shown in Fig. 18.

ACCURACY OF MEASUREMENTS

The main purpose behind the use of the measurements is the correct evaluation of the oscillograms. The voltage oscillograms, which were photographed about every two minutes, should give average curves for all the contact cycles. The judgment of the right time for the average oscillogram is a source of error. This error can be very considerable when cycles with different numbers of discharges are alternating. However, even the knowledge of the correct number of discharges is sufficient. It is known from the analysis of the current oscillograms that not every discharge causes removal of material. Hence, the accurate number of useful discharges is not known. It has been found that the specific material loss figures for a certain material under defined circuit conditions vary $\pm$ 20 per cent round an average value. This should indicate the accuracy of the investigation.

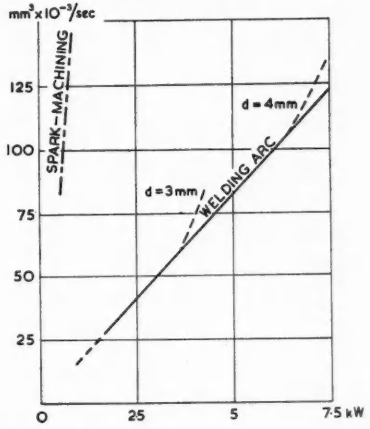


Fig. 17. **Volume transport curves for welding and spark-machining.**

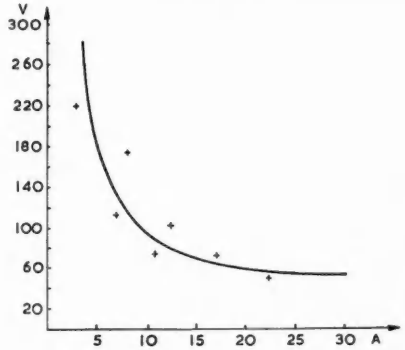


Fig. 18. **Minimum direct current required for arc formation.**

INDUSTRIAL POSSIBILITIES OF SPARK-MACHINING

In Fig. 19, spark-machining is compared to ordinary drilling, turning, etc. The material removal for different metals and for different kinds of machining operations is plotted against the power input. It is clear from this figure that spark machining is not economical for metals of medium hardness. It is, however, advantageous for very hard materials, e.g., Widia steel. The drilling of a hole of 0·16 in. dia. and 0·4 in. length in Widia steel takes 6-10 minutes.

The speed of spark machining can be considerably increased by the increase of the number of discharges. This is limited by the charging time, which is determined by the equation of the voltage increase on the condenser, $U = U_o\,(1 - e^{-\frac{t}{RC}})$. It is a matter of expediency at which value of U/U_o the discharge will begin. This equation is valid in cases where the self-inductance of the circuit is small and the circuit is aperiodic. However, by increasing the self-inductance of the circuit, the condenser charging becomes oscillatory with voltage peaks much higher than the voltage V of the generator. If the condenser is made to discharge at such a time, the efficiency and speed of the drilling

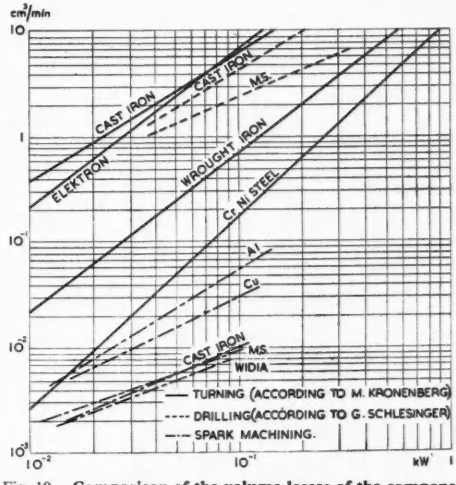


Fig. 19. **Comparison of the volume losses of the component by different machining methods.**

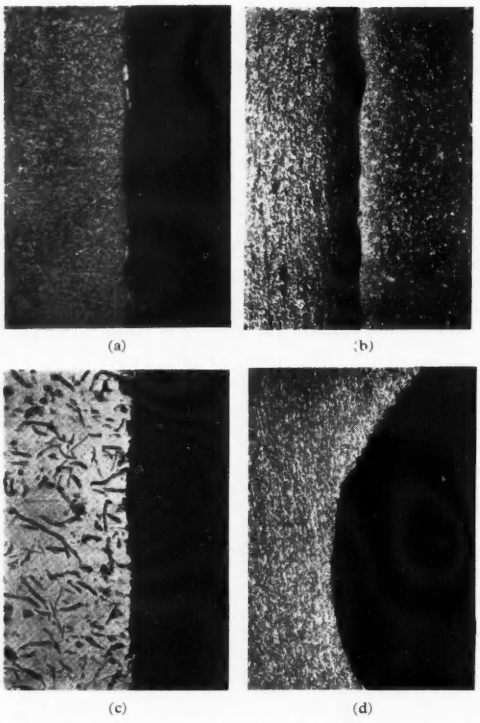


Fig. 20. **Micro-photographs of the drilled holes** (× 100 × 0·425).
(a) Spark-machining, mild steel ;
(b) Machine with the spiral drill, mild steel.;
(c) Spark-machining, cast iron ;
(d) Bottom end of the hole shown in (a).

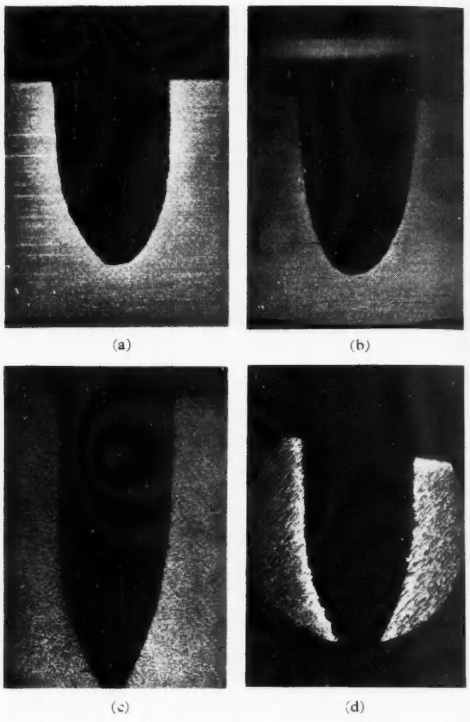


Fig. 21. **Macro-photographs of the cross-sections of drilled holes** (× 5 × 0·425).
(a) Mild steel ; (c) Cast iron ;
(b) Mild steel ; (d) Wildia.

process increases above that of the aperiodic circuit, because the voltage of the discharge is much higher and the voltage increase much steeper than in the aperiodic case. The arrangement of an oscillating charging circuit requires the correct timing of contact-making at a voltage peak on the condenser; this can be achieved by electronic means.

MICRO-PHOTOGRAPHS OF THE DRILLED HOLES

Figs. 20 and 21 show micro and macro-photographs of holes drilled by spark machining and Fig. 20b shows, for comparison, a hole drilled mechanically. It is clear that the boundary crystals are deformed in the mechanically drilled hole, but not in that made by spark-machining. No deep thermal effects can be detected in a spark-machined hole, and as the boundary crystals do not decrease in size, the temperature does not rise above its critical value. The surface of the spark-machined holes is very smooth. Fig. 20c shows a cross-section in cast iron, from which it is interesting to observe the influence of the discharge explosions on the boundary graphite inclusions.

The macro-photographs of Fig. 21 are enlarged 2·125 times. The uneven edges are a consequence of the cross-section grinding and not of the hole drilling.

CONCLUSION

The following are the main results of these investigations:

(1) The material removal of the machined component is proportional to the power taken from the condenser. The specified loss values in cu. in. × 10^{-9} /joule are:

Mild Steel	*Cast Iron*	*Widia Steel*	*Copper*	*Alum'um*
111	73	98	195	348

(2) The loss-per-discharge curves become straight lines above 100 V.

(3) The arc which may follow the discharge process is supplied by the direct current source. It is to be suppressed, as it decreases the efficiency of spark-machining.

(4) Spark-machining probably consists in the heating of the contact above boiling point and sudden disintegration. A subsequently formed arc does not increase the loss of material.

(5) Spark machining is not economical when compared to the usual methods of machining on medium hard metals. However, it becomes very advantageous from an economic standpoint and as regards quality of machining on very hard metals, and it is the only possible method, apart from diamond grinding, for the hardest metals.

(6) The efficiency of spark-machining is constant above 100 V.

(7) The quality of the drilled holes increases if the material loss/discharge decreases.

(8) By using an oscillating charging circuit, the voltage of the d.c. generator can be lowered by nearly 50 per cent. This increases the safety of the personnel.

The work described was carried out at the Mechanical Technological Institute of the Technical University in Budapest with the co-operation of the Research Section of the Centre for Heavy Industry.

Steam Jet Ejector Characteristics

By L. D. BERMAN. (From *Elektritsheskie Stantsii*, No. 7, 1949, pp. 17-22, 10 illustrations.)

THE vacuum p_2^I produced by a steam jet ejector depends upon the amount of air entering the steam condenser to which the ejector is connected and upon the conditions and composition of the steam-air mixture supplied to the ejector. Thus, if a change occurs in the amount of air entering the condenser, or in the temperature of the steam-air mixture, then the suction pressure or vacuum p_2^I will also vary. In Fig. 1 are charted test data which indicate the influence of the operating characteristics of the steam jet ejector plant upon condenser vacuum as observed in four different installations. This graph clearly shows that for identical percentages of air in the steam, widely different condenser vacua may obtain according to the characteristics of the steam jet ejector plant fitted in the particular case considered.

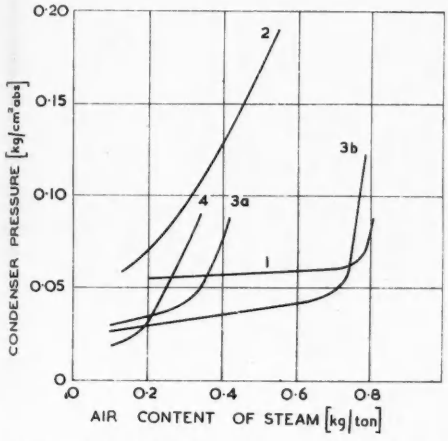


Fig. 1. **Variation in condenser vacuum with air content in steam for different steam jet ejector plants.**
(1) Condenser surface 1354 m².
(2) ,, ,, 1205 m².
(3) ,, ,, 1760 m².
(a) one group of ejectors in operation.
(b) two groups of ejectors in operation.
(4) Condenser surface 3116 m².

Curves (1) and (2) are based on atmospheric temperature of 25° C and condenser load of 30-35 kg of steam per hour per square metre condenser surface.

Curves (3) and (4) are based on atmospheric temperature of 1-3° C and condenser load of 36-38 kg/hr/m².

It is imperative that the turbine room operating personnel should be familiar with the operating characteristics of the steam jet ejector plant installed, and, in particular, with the influence of the various operational factors upon ejector vacuum. A fundamental ejector characteristic is the relationship prevailing between ejector vacuum and ejector load. Typical ejector characteristics as obtained on ejectors of various Russian makes are shown in Fig. 2, but it should be noted that these tests were carried out with air as aspirated medium instead of air-steam mixture. These characteristics show that, generally, an ejector characteristic consists of two distinct parts, namely, a gradually rising branch associated with operation in the design range and a steeply rising branch associated with overload conditions. As the characteristics shown in the graph are based on air, they do not give a full picture of the operational behaviour of an ejector plant in service where variations in the amount of steam present in the air-steam mixture affect ejector performance. With constant vacuum, an increase in the amount of steam

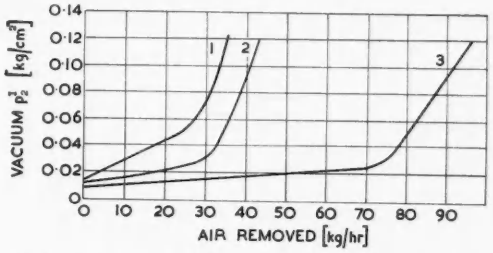


Fig. 2. **Test characteristics of various ejector plants obtained with aspiration of dry air.**
(1) Two-stage ejector; (2) Two-stage ejector; (3) Three-stage ejector.

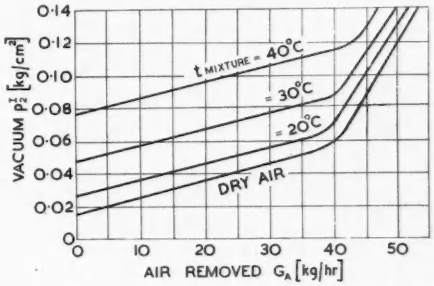


Fig. 3. **Ejector characteristics obtained with aspiration of steam-air mixture.**

present in the mixture will cause a diminution in ejector capacity owing to the decreased density of the mixture. For practical purposes, it is preferable, therefore, to relate the ejector vacuum, not to the weight of the mixture G_{Mi} kg per hour handled by the ejector, but to the amount of dry air G_A kg per hour removed by the ejector. A graph of this type is shown in Fig. 3 where ejector characteristics for dry air and for steam-air mixtures of 20, 30 and 40° C temperature are given.

First-stage ejector capacity for constant working steam pressure p_1 and a given vacuum p_2^I can, with sufficient accuracy, be assumed to depend only upon the air/steam weight ratio of the mixture, that is, on

$$\epsilon = \frac{G_A}{G_{Mi}} = \frac{p_2^I - p_n}{p_2^I - 0{\cdot}378\, p_n}$$

where p_n is the steam pressure associated with the mixture temperature t_{MI}. For values of $\epsilon \geqslant 0{\cdot}2$ the following empirical relationship was found to obtain over the design range :—

$$k_{Mi} = \frac{G_{MI}}{G_{A(d)}} = 0{\cdot}8 + 0{\cdot}2\,\epsilon$$

$$\text{or } k_A = \frac{G_A}{G_{A(d)}} = \frac{\epsilon\, G_{Mi}}{G_{A(d)}} = 0{\cdot}8\,\epsilon + 0{\cdot}2\,\epsilon^2$$

where $G_{A(d)}$ is the ejector capacity for the vacuum p_2^I when removing dry air at room temperature (15 - 20° C). A chart showing values for k_A for various mixture vacua and mixture temperatures of 20, 30, and 40° C is reproduced in Fig. 4. It should be noted that this chart solely applies to an ejector operating within the design range, and therefore does not apply to overload conditions.

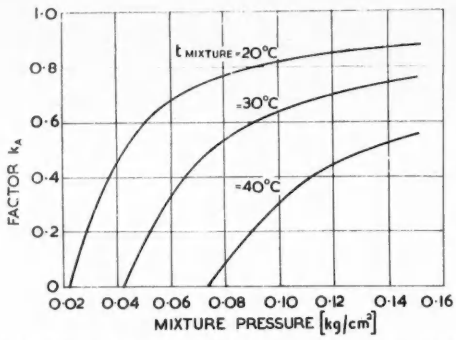


Fig. 4. **Factor k_A of ejector based upon aspiration of dry air.**

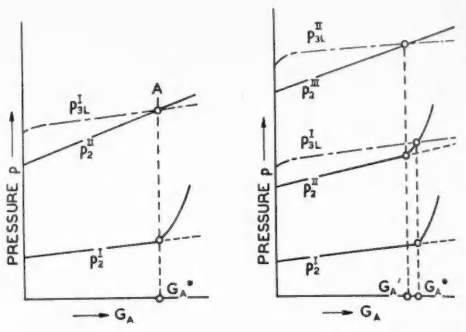


Fig. 5. **Individual stage characteristics.**
(a) Two-stage ejector; (b) Three-stage ejector.

It is known that for each stage of a steam jet ejector there exist two operating conditions, namely, one where the stage capacity for a given vacuum depends upon the back pressure, and the other where this is not the case. As long as the back pressure p_3 at the diffuser outlet does not exceed a given limiting value p_{3L}, the capacity of the ejector, based upon a given vaccum, p_2, will not vary with the back pressure p_3. But if the back pressure rises above the limiting value p_{3L}, then the ejector capacity falls sharply, or the vacuum sharply deteriorates.

In multi-stage ejectors the last stage operates with practically constant back pressure, as this is the atmospheric pressure, while the other stages operate against variable back pressures, the back pressure of one stage constituting the aspirating pressure of the subsequent stage. Thus, in the case of a two-stage ejector, the back pressure p_3^{I} of the first stage is identical with the vacuum or aspirating pressure of the second stage p_2^{II}. If the latter is less than the critical pressure p_{3L}^{I} in the case of varying G_A, both ejector stages will operate according to their normal operating characteristics, and the first stage will not influence the aspirating pressure of the second stage. However, if p_2^{II} increases to a point where it equals p_{3L}^{I} (point A in Fig. 5a), then a further increase in G_A will lead to a sharp increase in the aspirating pressure of the first stage, the ejector being overloaded.

Overloading of a three-stage ejector is produced in a similar manner and takes place if the inlet pressure of the second stage exceeds the limiting back pressure of the first stage. In this case, the second stage may either continue to operate within its normal operating range, with the pressure p_2^{II} being greater than the pressure p_{3L}^{I}, or the inlet pressure of the third stage may already have exceeded the limiting back pressure of the second stage. In the latter case, a sharp rise in the inlet pressure of the second stage will at once lead to overloading of the ejector (Fig. 5b). Therefore, the maximum operational output G_A^* of the ejector which must not be exceeded if overloading and consequent deterioration of the condenser vacuum is to be avoided, depends upon the operating characteristics of the individual ejector stages as well as upon the respective limiting back pressures.

For a given ejector capacity G_A, the limiting back pressure is generally dependent upon (a) ejector dimensions, (b) working steam flow rate, and (c) composition of the aspirated steam-air mixture. If all other conditions remain equal, a decrease in the ratio ϵ leads to an increase in p_{3L}^{I} for the same rate of G_A. Therefore, an increase in the temperatures of the aspirated mixture which stipulates a higher aspirating pressure increases somewhat the maximum operating capacity of the ejector, as shown in Fig. 3. However, this holds true only if the temperature of the water supplied to the interstage coolers of the ejector remains constant.

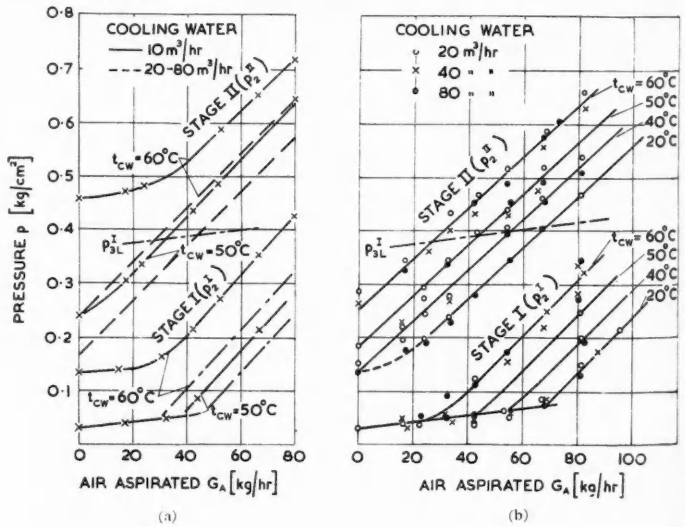


Fig. 6. **Ejector characteristics for variable ejector loads and cooling water temperatures t_{CW}.**

In actual fact, an increase in the temperature of the steam-air mixture caused by either an increase in condenser load or a rise in the temperature of the condenser cooling water is always accompanied by a rise in the temperature of the condensate passed into the ejector intercoolers, thus diminishing the limit capacity G_A^*, which must not be exceeded if overloading of the ejector is to be avoided.

The influence of variations in quantity and temperature of the cooling water supplied to the interstage coolers upon the ejector characteristics is exemplified by the graphs given in Figs. 6a and 6b. These data were obtained from tests conducted at the Steam Turbine Laboratory of the Leningrad Metal Works "Stalin" in 1941, the tests being conducted with dry air. These charts also show the slightly rising limiting back pressure characteristics p_{2L}^{I}. For a water temperature of 60° C, the characteristic of the second stage with a flow of cooling water of 10 m³/hr is seen in Fig. 6a to lie above the limiting back pressure curve of the first stage, with a corresponding flattening-out of the characteristics p_2^{I} and p_2^{II}, while in Fig. 6b a diminution in the flow rate from 80 to 20 m³/hr remains practically without influence upon the p_2^{II} characteristics.

An increase in cooling water temperature has a considerable effect upon the ejector characteristics by limiting its maximum capacity. As will be seen by reference to Fig. 6b, the ejector is capable of handling up to 65 kg/hr of air when supplied with 20 m³/hr of water at 20° C, but if the water temperature is increased to 60° C, then the maximum load drops to about 25 kg/hr (Fig. 6a).

Change Wheel Gearing

By O. LICHTWITZ, M.I.MECH.E.

(*Continued from April issue*).

CHAPTER VII

GEARING OF MACHINE TOOLS

Machine tool makers simplify change gear problems as far as possible by supplying charts; in some cases, however, the only instructions are formulae for the ratio. The formulae for computing the gear ratios should be known in any case.

Screw cutting lathes are equipped with rather simple charts which are sufficient for the common standardised threads; any pitch not contained in the charts calls for calculating a gear train.

The milling machine operator has a more extensive list of leads, which are obtainable by means of the usual set of change gears, so that the nearest to any required value can readily be chosen. In some cases, however, special gear trains must be computed to meet particular requirements.

Other machine tools require a practically unlimited number of different gear trains, and the ratio must in every case be very close to a required value. The cutting of high-quality helical gears involves a multiplicity of gear combinations, whether they are hobbed differentially or non-differentially, or produced by any other process. In such cases, charts simplify the computing of gear trains, but do not replace it entirely.

(a) *Gearing of the Lathe*

The lathe is the most familiar machine tool to the engineer. It is true that the lathe has lost much in importance, and it has been supplanted either by automatic screw machines, thread rolling, thread milling, thread grinding, worm milling, and similar machines for the mass production of threads. The connection of the rotary movement of the work with the longitudinal movement of the tool or the work, by means of change gears, is the same for many of these machines as for the lathe.

Most of the modern lathes are equipped with a quick change gear box, which provides the proper gearing for the most usual pitches by a mere manipulation of levers. These lathes are usually equipped with only a few change gears, which, even in conjunction with the gear box, are less versatile than the ordinary screw cutting lathe with a large number of change gears; the following considerations refer to the ordinary type of lathe.

It is assumed that readers are familiar with the lathe and the principles of screw cutting. It should merely be mentioned that nearly all lathes are geared so that if equal gears are placed on the spindle stud and on the lead screw, a thread is produced with the same pitch as that of the lead screw. Should the lathe be different, the pitch which is cut by equal gears may be regarded as the pitch of the lead screw.

If the lead screw has a pitch of $\frac{1}{4}$ in. (or 4 threads per inch), and if a thread with a pitch of $\frac{1}{2}$ in. is to be cut, the lead screw must rotate twice as fast as the work. That is achieved by gears with the ratio 2/1, the larger driving the smaller. If the pitch to be cut is $\frac{1}{8}$ in., the lead screw must rotate with half the speed of the spindle; gears with the ratio 1/2 would be used, the smaller driving the larger.

Generally, the ratio $\frac{\text{driving gears}}{\text{driven gears}} = \frac{\text{pitch to be cut}}{\text{pitch of lead screw}}$

or $\frac{\text{t.p.i. of lead screw}}{\text{t.p.i. to be cut}}$.

The change gears of most lathes have multiples of 4, 5, or 6 as number of teeth.

The lead screws of many lathes have $\frac{1}{2}$ in. or $\frac{1}{4}$ in. pitch, but also 6, 8, 10, or 12 threads per inch are used on smaller lathes.

Many tables would be necessary to cover the diversity of lathes and change gears. If, however, a gear train is known for one machine, a gear train for a different machine can mostly be obtained by a simple transformation. If, for example, the gear train for a lathe with $\frac{1}{2}$ in. lead screw is $\frac{50 \times 35}{65 \times 120}$, a lathe with a lead screw with 6 t.p.i. requires the 3-fold ratio, that is $\frac{105 \times 50}{65 \times 120}$. If this lathe should be equipped with change gears in steps of 4, the gear train can be transposed $\frac{105 \times 50}{65 \times 120} = \frac{5 \times 7}{4 \times 13} = \frac{40 \times 28}{32 \times 52}$, or if the gears are in steps of 6, $\frac{5 \times 7}{4 \times 13} = \frac{60 \times 42}{48 \times 78}$, or if they are mixed of multiples of 4 and 6, $(5 \times 7)/(4 \times 13) = (42 \times 30)/(36 \times 52)$.

Table II contains the gear trains for cutting any British and U.S.A. standard thread (with the exception of B.A. threads) on lathes with $\frac{1}{2}$ in. and $\frac{1}{4}$ in. lead screws. The gears should as far as possible be mounted in the order indicated in Table II (and the following tables). The finest threads, on a lathe with $\frac{1}{2}$ in. lead screw can only be cut by means of 6-gear trains.

TABLE II. GEAR TRAINS FOR CUTTING STANDARD THREADS.

Threads per inch	Lead screw ½ in. pitch		Lead screw ¼ in. pitch	
	Driving gears	Driven gears	Driving gears	Driven gears
2½	120.70	100.105	120.80	60.100
2⅝	120.60	90.105	120.100	75.105
2¾	120.60	90.110	120.100	75.110
2⅞*	120.70	105.115	120.100	75.115
3	110.40	55.120	120	90
3¼	120.35	65.105	120.70	65.105
3½	120.55	105.110	120	105
4	110.30	55.120	120.55	60.110
4½	120.35	90.105	120.70	90.105
5	120.35	100.105	120.70	100.105
6	110.20	55.120	110.40	55.120
7	90.40	105.120	120.55	105.110
8	105.20	70.120	110.30	55.120
9	100.20	75.120	120.35	90.105
10	80.30	100.120	120.35	100.105
11	80.30	110.120	120.35	105.110
12	80.25	100.120	110.20	55.120
13	60.20	65.120	80.30	65.120
14	90.20	105.120	90.40	105.120
16	75.20	100.120	105.20	70.120
18	70.20	105.120	100.20	75.120
19	60.20	95.120	80.30	95.120
20	60.20	100.120	80.30	100.120
22	60.20	110.120	80.30	110.120
24	55.20	110.120	80.25	100.120
26	65.20	30.120	60.20	65.120
28	45.20	105.120	90.20	105.120
32	30.25	100.120	75.20	100.120
36	35.20	105.120	70.20	105.120
40	30.20	100.120	60.20	100.120
44	30.20	110.120	60.20	110.120
48	25.20	100.120	55.20	110.120
56	90.25.20	100.105.120	45.20	105.120
64	75.20.20	80.100.120	30.25	100.120
72	75.20.20	90.100.120	35.20	105.120
80	60.20.20	80.100.120	30.20	100.120
2⅞	90.85	100.110	90.85	50.110

*The gear 115 can be avoided by the substitute gear train (error 0·28 per 1000).

Example 24. To cut on a lathe having a lead screw with 2 t.p.i.,
(a) 11 t.p.i., and (b) 4½ t.p.i.;

(a) The ratio $\frac{\text{driving gears}}{\text{driven gears}} = \frac{2}{11}$. The simple gear train $\frac{20}{110}$, or the compound gear train $\frac{80\times 30}{110\times 120}$ are suitable.

(b) The ratio $\frac{\text{driving gears}}{\text{driven gears}} = \frac{2}{4\frac{1}{2}} = \frac{4}{9}$. The simple gear train 20/45 or the more favourable 40/90 are suitable; the four-gear train $\frac{120\times 35}{90\times 105}$ places still larger gears on the spindle stud and lead screw.

As Example 24 shows, fractional pitches such as 4½ t.p.i., do not involve problems other than, for example, 11 t.p.i. However, in some cases they require unusual gears. In order to cut 2⅞ t.p.i. on a lathe with ⅛ in. pitch lead screw, the ratio $2/2\frac{7}{8} = 16/23$ is required; a gear with a multiple of 23 teeth is necessary for that purpose.

The rather uncommon Whitworth threads 4¼ in. or 4½ in. are the only cases where the gear 115 (= 5 × 23) is required. That gear can be dispensed with the more as the gear train $\frac{90\times 85}{100\times 110}$ enables cutting of $\frac{100\times 110}{90\times 85}\times 2 = 2{\cdot}8758$ t.p.i., a fairly close approximation to the required 2·875 t.p.i. The accumulated error on a 3½ in. long thread would only be 0·001 in.

Metric threads are indicated by the pitch in millimetres. As 1 inch is equal to 25·4 mm (the legal equivalent of 1 inch in Great Britain is insignificantly less, in the U.S.A. insignificantly greater than 25·4 mm), 1 mm = 1/25·4 = 5/127 in.

The ratio for cutting a pitch of, for instance, 2 mm on a lathe with ¼ in. lead screw is $\frac{\text{pitch to be cut}}{\text{pitch of lead screw}} = \frac{2\times 5}{127}\div\frac{1}{4} = \frac{2\times 5\times 4}{127} = \frac{40}{127}$.

The translating gear 127 is a convenient means for cutting metric threads; many metric threads can be cut without that gear if a small error is permissible; some approximations for 25·4 are

(1) $(40\times 40)/63 = 25{\cdot}39683$; error 0·12 per 1,000
(2) $(18\times 24)/17 = 25{\cdot}41176$; error 0·46 per 1,000
(3) $(15\times 22)/13 = 25{\cdot}38461$; error 0·61 per 1,000

These ratios, substituted for 25·4, lead in many cases to suitable gear trains. In Table III the gear train for 5 mm is based on the ratio (1), for 3 mm on the ratio (2), or for 1·5 mm (lead screw ½ in.) on the ratio (3).

Some lathes are equipped for the cutting of metric threads with a gear 63; the gear trains on the charts, however, often are not based on the ratio $(40\times 40)/63$ but on $(2\times 63)/5 = 25{\cdot}2$, with an error of 7·87 per 1,000. If, for example, a pitch of 4·5 mm is to be cut on a lathe with ¼ in. lead screw, where the ratio 18/25·4 is required, the gear train $18\div(40\times 40)/63 = (90\times 63)/(80\times 100)$ is a good approximation; the gear train $18\div(2\times 63)/5 = 45/63$ may be found on a chart, but it is inferior. Turners should be cautious if they find 63 as one of the driven gears.

The errors for the finer pitches in Table III are rather high, and the finest pitches are not obtainable at all on lathes with ½ in. lead screw, even by means of compound gear trains. Table IV enables better results for these pitches, if the lathe is equipped for six-gear trains.

If a metric thread which is not shown in Table III, is to be cut, a gear train can in many cases be obtained from that table. If, for instance, a thread with 12 mm pitch is to be cut in a lathe with a ½ in. lead screw, the ratio is twice that required for 6 mm pitch, that is $2\times\frac{120\times 55}{110\times 127} = \frac{120}{127}$.

If a gear train is derived in this way from a train which involves an error, it is affected by the same relative error. If, for example, the gear train for 12 mm pitch is derived from the gear train $\frac{85\times 60}{90\times 20}$ for 6 mm (Table III) the gear train $2\times\frac{85\times 60}{90\times 120} = \frac{85}{90}$ involves an error of 0·46 per 1,000. By computing a gear train as demonstrated in Chapter III, the better approximation $(105\times 45)/(50\times 100)$ is obtained, with an error of only 0·19 per 1,000. This gear train is identical with that shown in Table III for cutting 6 mm pitch on a lathe with ¼ in. lead screw.

If a screw is to be replaced, the pitch can be ascertained by a screw pitch gauge, or by directly counting

TABLE III. GEAR TRAINS FOR CUTTING METRIC THREADS.

Pitch in mm	Pitch in inches	Threads per inch	Lead screw ¼ in. pitch				Lead screw ⅛ in. pitch			
			Driving gears	Driven gears	Special gear	Error per 1,000	Driving gears	Driven gears	Special gear	Error per 1,000
0·25	0·0098	101·60					20.20	85.120		3·92
							25.20	100.127	127	0
0·3	0·0118	84·67					20.20	85.100		3·92
							30.20	100.127	127	0
0·35	0·0138	72·57					35.20	105.120		7·94
							35.20	100.127	127	0
0·4	0·0157	63·50	20.20	105.120		7·94	30.20	95.100		2·63
			20.20	100.127	127	0	40.20	100.127	127	0
0·45	0·0177	56·44	20.20	95.120		9·75	35.20	90.110		2·24
							45.20	100.127	127	0
0·5	0·0197	50·80	20.20	85.120		3·92	45.20	95.120		2·63
			25.20	100.127	127	0	60.20	120.127	127	0
0·6	0·0236	42·33	20.20	85.100		3·92	50.25	110.120		2·21
			30.20	100.127	127	0	60.20	100.127	127	0
0·7	0·0276	36·29	35.20	105.120		7·94	55.20	95.105		0·36
			35.20	100.127	127	0	70.20	100.127	127	0
0·75	0·0295	33·87	20·20	75.90		3·46	65.20	100.110		0·61
			30.25	100.127	127	0	90.20	120.127	127	0
0·8	0·0315	31·75	30.20	95.100		2·63	30.25	70.85		0·53
			40.20	100.127	127	0	80.20	100.127	127	0
0·9	0·0354	28·22	35·20	90.110		2·24	35.25	65.95		0.22
			45.20	100.127	127	0	90.20	100.127	127	0
1	0·0394	25·40	45.20	95.120		2·63	85.20	90.120		0·46
			60.20	120.127	127	0	80.30	120.127	127	0
1·25	0·0492	20·32	65.20	110.120		0·61	45.35	80.100		0·12
			75.20	120.127	127	0	100.30	120.127	127	0
1·5	0·0591	16·93	65.20	100.110		0·61	85.30	90.120		0·46
			90.20	120.127	127	0	100 25	100.127	127	0
1·75	0·0689	14·51	55.25	95.105		0·36	55.50	95.105		0·36
			105.20	120.127	127	0	105.40	120.127	127	0
2	0·0787	12·70	85.20	90.120		0·46	45.35	50.100		0·12
			80.30	120.127	127	0	120.35	105.127	127	0
2·5	0·0984	10·16	45·35	80.100		0·12	105.45	100.120		0·12
			100·30	120.127	127	0	120.25	60.127	127	0
3	0·1181	8·47	85.30	90.120		0·46	85.60	90.120		0·46
			120.25	100.127	127	0	120.55	110.127	127	0
3·5	0·1378	7·26	55.50	95.105		0·36	90.45	70.105		0·29
			105.40	120.127	127	0	120.35	60.127	127	0
4	0·1575	6·35	45.35	50.100		0·12	105.45	75.100		0·12
			120.35	105.127	127	0	120.70	105.127	127	0
4·5	0·1772	5·64	85.45	90.120		0·46	85	120		0·46
			120.30	80.127	127	0	120.75	100.127	127	0
5	0·1969	5·08	105.45	100.120		0·12	105.90	100.120		0·12
			120.25	60.127	127	0	120.75	90.127	127	0
5·5	0·2165	4·62	90.45	85.110		0·19	90.45	55 85		0·19
			110.60	120.127	127	0	120.55	60.127	127	0
6	0·2362	4·23	85.60	90.120		0·46	105.45	50.100		0·19
			120.55	110.127	127	0	120	127	127	0

the threads per inch. In the case of a worm the pitch must be carefully measured. Some sizes differ by very little only; for instance, the difference in the pitch between c.p. $\frac{5}{16}$ in. and d.p. 10 is less than 0·002 in. If it is possible to measure a length of several pitches, the margin of errors in measurement is reduced; if the meshing worm wheel is available, the pitch can be ascertained still more accurately.

Many worms have more than one start or thread, and the difference between pitch and lead must be distinguished; the pitch is the distance between the threads or teeth in the axial direction; the lead is the distance the thread advances in one turn. The lead of a two- or three-start worm is, therefore, twice or three times the pitch.

Table V contains the gear trains for worms likely to be met with in practice. Worm gears are often based on the circular pitch, because of the simplicity and accuracy of the gear trains for cutting the worm.

Example 25. To cut a three-start worm, $\frac{7}{8}$ in. circular pitch, in a lathe with $\frac{1}{2}$ in. lead screw.

The lead is $3 \times (7/8) = 21/8 = 2\frac{5}{8}$ in. The required ratio is $2\frac{5}{8}/\frac{1}{2} = 21/4$.

The simple train 105/20, or a more favourable com-

TABLE IV. SIX-GEAR TRAINS FOR CUTTING METRIC THREADS.

Pitch in mm	Lead screw ¼ in. pitch				Lead screw ⅛ in. pitch			
	Driving gears	Driven gears	Special gear	Error per 1,000	Driving gears	Driven gears	Special gear	Error per 1,000
0·25	65·20.20	100·110·120		0·61	55·25·20	70·95.105		0·36
	75.20.20	100.120.127	127	0				
0·30	35.25.20	65.95.120		0·22	70.25.20	65.95.120		0.22
	90.20.20	100.120.127	127	0				
0·35	55.25.20	95.100.105		0·36	110.25.20	95.100.105		0·36
	105.20.20	100.120.127	127	0				
0·4	35.25.20	65.90.95		0·22	70.25.20	65.90.95		0·22
0·45	35.30.25	65.95.120		0·22	95.20.20	65.75.110		0·05
	90.30.20	100.120.127	127	0				
0·5	55.25.20	70.95.105		0·36	45.35.30	50.100.120		0·12
0·6	70.25.20	65.95.120		0·22	70.50.20	65.95.120		0·22
0·7	110.25.20	95.100.105		0·36	100.50.25	90.105.120		0·06
0·75	95.20.20	65.90.110		0·05	95.40.20	65.90.110		0·05
0·8	70.25.20	65.90.95		0·22	45.35.30	50.75.100		0·12
0·9	95.20.20	65.75.110		0·05	95.40.20	65.75.110		0·05
1	45.35.30	50.100.120		0·12	105.45.30	75.100.120		0·12

TABLE V. GEAR TRAINS FOR CUTTING WORM THREADS

Circular pitch	Diametral pitch	Module	Pitch in inches	Threads per inch	Lead screw ½ in. pitch				Lead screw ¼ in. pitch			
					Driving gears	Driven gears	Special gear	Error per 1,000	Driving gears	Driven gears	Special gear	Error per 1,000
		0·5	0·0618	16·17	65.20 55.30 60.20	100.105 105.127 97.100	127 97	1·01 0·40 0·21	95.25	80.120		0.11
1/16			0·0625	16·00	75 20	100.120		0	105.20	70.120		0
	48		0·0654	15·28	55.30	105.120		0·40	110.30 95.35 55.30	105.120 100.127 65.97	127 97	0.40 0·04 0·39
	40		0·0785	12·73	55.30 80.20	100.105 97.105	97	0·40 0·09	70.35 97.20	65.120 65.95	97	0·18 0.03
	36		0·0873	11·46	110.20 55.20	105.120 65.97	97	0.40 0·39	110.40 95.35 110.20	105.120 75.127 65.97	127 97	0·40 0 04 0·39
		0·75	0·0928	10·78	70.35 55 45 90.20	110.120 105.127 97.100	127 97	0.43 0·40 0·21	85.55	105.120		0·06
	32		0·0982	10·19	55.45 100.20	105.120 97.105	97	0·40 0·09	95 25 97.25	55.110 65.95	97	0.35 0·03
	30		0·1047	9·55	110.20 65.25	100.105 80.97	97	0·40 0·15	70.35 75.65	65.90 97.120	97	0·18 0·15
	28		0·1122	8·91	70.25 95.30	65.120 100.127	127	0·18 0·04	100.35 95.60	65.120 100.127	127	0·18 0·04
	26		0·1208	8·28	55.30 75.45 75.25	65.105 110.127 80.97	127 97	0·40 0·30 0·15	65.45 90.75 75.50	55.110 110.127 80.97	127 97	0·31 0·30 0·15
		1	0·1237	8·09	95 25	80.120		0·11	85.55	90.105		0·06
1/8			0·1250	8·00	105.20	70.120		0	110.30	55.120		0
	24		0·1309	7·64	110.30 95.35 55.30	105.120 100.127 65.97	127 97	0·40 0·04 0·39	110.60 95.70 110.30	105.120 100.127 65.97	127 97	0·40 0·04 0·39
	22		0.1428	7.10	90.40 95.35	105.120 97.120	97	0.40 0·19	120.55 95.70	105.110 97.120	97	0·40 0·19
		1·25	0·1546	6·47	85.40 120.25	100.110 97.100	97	0·39 0·21	85.80 120.55 97·85	100.110 97.110 105.127	97 97, 127	0·39 0·21 0·20
	20		0·1571	6·37	70.35 97.20	65.120 65.95	97	0·18 0.03	105.35 97.40	65.90 65.95	97	0,18 0·03
	18		0·1745	5·73	110.40 95·35 110.20	105.120 75.127 65.97	127 97	0.40 0·04 0·39	110.80 95.70 97.95	105.120 75.127 110.120	127 97	0·40 0·044 0·037
		1·5	0·1855	5·39	85.55	105.120		0·06	110.85	105.120		0·06
3/16			0·1875	5·33	105·30	70.120		0	110.45	55.120		0
	16		0.1963	5·09	95.25 97.25	55.110 65.95	97	0·35 0·03	95·50 105.95 97.50	55.110 100.127 65.95	127 97	0·35 0·04 0·03
	15		0·2094	4·77	70.35 75.65	65.90 97.120	97	0·18 0·15	70.35 100.65	45.65 80.97	97	0·18 0·15
		1·75	0·2164	4·62	100.50	105.110		0·01	100.50	55.105		0·01
	14		0·2244	4·46	100.35 95.60	65.120 100.127	127	0·18 0·04	105.50 120.95	65.90 100.127	127	0·18 0·04
	13		0·2417	4·14	65.45 90.75 75·50	55.110 110.127 80·97	127 97	0·31 0·30 0·15	90.65 90.75 100.75	55.110 55.127 80·97	127 97	0·31 0·30 0·15
		2	0·2474	4·04	85.55	90.105		0·06	110·85	90.105		0·06
1/4			0·2500	4·00	110.30	55.120		0	110.60	55.120		0
	12		0·2618	3·82	110.60 95.70 110.30	105.120 100.127 65.97	127 97	0·40 0·04 0·39	110.80 105.95 97.95	70.120 75.127 80.110	127 97	0·40 0·044 0·037
		2·25	0·2783	3·59	85·55	70.120		0·06	110.85	70.120		0·06
	11		0·2856	3·50	120.55 95.70	105.110 97.120	97	0·40 0·19	120 105.95	105 90.97	97	0·40 0·19
		2·5	0·3092	3·23	85·80 120.55 97.85	100.110 97.110 105.127	97 97, 127	0·39 0·21 0·20	120·85 127.75 120	75·110 70.110 97	127 97	0·39 0·13 0·21
5/16			0·3125	3·20	105.50	70.120		0	105.100	70.120		0
	10		0·3142	3·18	105.35 97.40	65.90 65.95	97	0.18 0·03	105.70 127·95 97.80	65.90 80.120 65.95	127 97	0·18 0·11 0·03
		2·75	0·3401	2·94	100.50	70.105		0·01	100.50	35.105		0·01
	9		0·3491	2·86	110.80 95.70 97.95	105.120 75.127 110.120	127 97	0·40 0·044 0·037	120.110 127.80	90.105 75.97	97, 127	0·40 0·21
		3	0·3711	2·69	110.85	105.120		0·06	110.85	60.105		0·06
3/8			0·3750	2·67	110.45	55.120		0	120.55	40.110		0
	8		0·3927	2·55	95.50 105.95 97.50	55.110 100.127 65.95	127 97	0·35 0·04 0·03	100.95 105.95 100.97	55.110 50.127 65.95	127 97	0·35 0·04 0.03
		3·25	0·4020	2·49	95.55 100.97	65.100 95.127	97,127	0·13 0·03	110.95	65.100		0·13
		3·5	0·4329	2·31	100.50	55.105		0·01	110.85	45.120		0·06
7/16			0·4375	2·29	105	120		0	110.105	55.120		0
	7		0·4488	2 23	105.50 120.95	65.90 100.127	127	0·18 0·04	105.100 120.95	65.90 50.127	127	0·18 0·04
		3·75	0·4638	2·16	120.85 120.75 97.85	100.110 97.100 70.127	97 97, 127	0·39 0·21 0·20	120.85 120.105 97.85	50.110 70.97 35.127	97 97, 127	0·39 0·21 0·20

TABLE V.—*Continued.*

TABLE V.—*Continued.*

Circular pitch	Diametral pitch	Module	Pitch in inches	Threads per inch	Lead screw ½ in. pitch				Lead screw ¼ in. pitch			
					Driving gears	Driven gears	Special gear	Error per 1,000	Driving gears	Driven gears	Special gear	Error per 1,000
		4	0·4947	2·02	110.85	90.105		0·06	110·85	45.105		0·06
1/2			0·5000	2.00	110.60	55.120		0	105.80	35.120		0
	6		0·5236	1·91	110.80 105.95 97.95	70.120 75.127 80.110	127 97	0·40 0·044 0·037	110.80 95.70 97.95	35.120 25.127 40.110	127 97	0·40 0·044 0·037
		4·5	0·5566	1·80	110.85	70.120		0·06	110.85	35.120		0·06
9/16			0·5625	1·78	105.90	70.120		0	105.90	35.120		0
		5	0·6184	1·62	120.85 127.75 120	75.110 70.110 97	127 97	0·39 0·13 0·21	85.80 127.75 120.110	25.110 70.110 55.97	127 97	0·39 0·13 0·21
5/8			0·6250	1·60	105.100	70.120		0	105.100	35.120		0
	5		0·6283	1·59	105.70 127.95 97.80	65.90 80.120 65.95	127 97	0·18 0·11 0·03	105.70 127.95 75.65	45.65 40.120 20.97	127 97	0·18 0.11 0·15
		5·5	0·6803	1·47	100.50	35.105		0·01	85.80 127.90 120.110	25.100 35.120 50.97	127 97	0·39 0·13 0·21
11/16			0·6875	1·45	110.105	70.120		0	110.105	35.120		0
		6	0·7421	1·35	110.85	60.105		0·06	110.85	30.105		0·06
3/4			0·7500	1·33	120.55	40.110		0	120.55	20.110		0
	4		0·7854	1·27	100.95 105.95 100.97	55.110 50.127 65.95	127 97	0·35 0·04 0·03	120.110 105.95	40.105 25.127	127	0·40 0·04
13/16			0·8125	1·23	105.65	35.120		0	120.65	30.80		0
		7	0·8658	1·16	110.85	45.120		0·06	110.85	30.90		0·06
7/8			0·8750	1·14	110.105	55.120		0	105.100	25.120		0
	3½		0·8976	1·11	105.100 120.95	65.90 50.127	127	0·18 0·04	105.100 120.95	45.65 25.127		0·18 0·04
15/16			0·9375	1·07	105.75	35.120		0	100.90	20.120		0
		8	0·9895	1·01	110.85	45.105		0·06	100.95	20.120		0·11
1			1·0000	1·00	105.80	35.120		0	120.70	20.105		0
	3		1 0472	0·955	110.80 95.70 97.95	35.120 25.127 40.110	127 97	0·40 0·044 0·037	120.110 97.95	30.105 20.110	97	0·40 0·037
		9	1·1131	0·898	110.85	35.120		0·06	110.85	20.105		0·06
1¼			1·1250	0·889	105·90	35.120		0	120·75	20·100		0
	2¾		1·1424	0·875	120.110 127.85 105.95	55.105 45.105 45.97	127 97	0·40 0·06 0·19	120.100	25.105		0·40

Circular pitch	Diametral pitch	Module	Pitch in inches	Threads per inch	Lead screw ½ in. pitch				Lead screw ¼ in. pitch			
					Driving gears	Driven gears	Special gear	Error per 1,000	Driving gears	Driven gears	Special gear	Error per 1,000
		10	1·2368	0·809	85.80 127.75 120.110	25.110 55.70 55.97	127 97	0·39 0·13 0·21	85.80 127.75 120.100	25.55 35.55 25.97	127 97	0·39 0·13 0·21
1¼			1·2500	0·800	105.100	35.120		0	120.100	30.80		0
	2½		1·2566	0·796	105.70 127.95 75.65	45.65 40.120 20.97	127 97	0·18 0.11 0·15	120.110 127.95	25.105 20.120	127	0·40 0·11
		11	1·3606	0·735	85.80 127.90 120·110	25.100 35.120 50.97	127 97	0·39 0·13 0·21	120.85 127.90 120.110	25.75 20.105 25.97	127 97	0·39 0·13 0·21
1⅜			1·3750	0·727	110.105	35.120		0	120.110	30.80		0
	2¾		1·3963	0·716	120.110 97.95	45.105 30.110	97	0·40 0·04	110.80 97.95	35.45 30.55	97	0·40 0·04
		12	1·4842	0·674	110.85	30·105		0·06	110.85	35·45		0·06
1½			1·5000	0·667	120.55	20.110		0	120.100	25.80		0
	2		1·5708	0·637	120.110 105.95	40.105 25.127	127	0·40 0·04	120.110 127.120	30.70 25.97	97, 127	0·40 0·21
		13	1·6079	0·622	110.95	50.65		0·13	110.95	25.65		0·13
1⅝			1·6250	0·615	120.65	30.80		0	120.65	20.60		0
		14	1·7316	0·578	110.85	30.90		0·06	110.85	30.45		0·06
1¾			1·7500	0·571	105.100	25.120		0	120.105	20.90		0
	1¾		1·7952	0·557	105.100 120.95	45.65 25.127	127	0·18 0·04	95.85 100.97	25.45 30.45	97	0·86 0·61
		15	1·8553	0·539	120.85 120.75	25.110 25.97	97	0·39 0·21	120.85	25.55		0·39
1⅞			1·8750	0·533	100.90	20.120		0	120.100	20.80		0
		16	1·9789	0·505	100.95	20·120		0·11	100.95	20.60		0.11
2			2·0000	0·500	120.70	20.105		0	120.100	20.75		0
	1½		2·0944	0·477	120.110 97.95	30.105 20.110	97	0·40 0 04	120.110 97.95	35.45 20.55	97	0·40 0·04
2¼			2·2500	0·444	120.75	20.100		0	120.105	20.70		0
2½			2·5000	0·400	120.100	30.80		0	120.100	20.60		0
	1¼		2·5133	0·398	120.110 127.95	25.105 20.120	127	0·40 0.11	110·80 127.95	25.35 20.60	127	0·40 0·11
2¾			2·7500	0·364	120.110	30.80		0	120.110	20.60		0
3			3·0000	0·333	120.100	25.80		0	120.110	20.55		0
	1		3·1416	0·318	120.110 127.120	30.70 25.97	97,127	0.40 0·21	120.110	30.35		0·40

pound train, for instance $(120 \times 105)/(30 \times 80)$ is suitable. Both the gears 105 and 120 as first drivers enable indexing of 3. The same result can also be obtained by multiplying the gear train $(110 \times 105)/(55 \times 120)$ (for c.p. $\frac{3}{8}$ in.) in Table V., by 3.

The diametral pitch system, which in spur, helical, and bevel gears is predominant, is also extensively used for worm gearing. As the value π cannot be exactly expressed by the quotient of two numbers, the circular pitch $P' = \pi/P$ (P = diametral pitch) can be obtained approximately only; the errors can, however, be kept negligible.

Some values near to $\pi = 3{\cdot}1415927$ are

(1) $\dfrac{5 \times 71}{113} = 3{\cdot}1415929$; error 0·00 per 1,000

(2) $\dfrac{13 \times 29}{4 \times 30} = 3{\cdot}1416667$; error 0·02 per 1,000.

(3) $\dfrac{8 \times 97}{13 \times 19} = 3{\cdot}1417004$; error 0·037 per 1,000.

(4) $\dfrac{19 \times 21}{127} = 3{\cdot}1417323$; error 0·044 per 1,000.

(5) $\dfrac{5 \times 49}{6 \times 13} = 3{\cdot}1410256$; error 0·18 per 1,000.

(6) $\dfrac{39 \times 39}{22 \times 22} = 3{\cdot}1425620$; error 0·31 per 1,000.

(7) $\dfrac{4 \times 95}{11 \times 11} = 3{\cdot}1404959$; error 0·35 per 1,000.

(8) $\dfrac{22}{7} = 3{\cdot}1428571$; error 0·40 per 1,000.

The first value, with practically no error, requires two special gears, 71 and 113, which are justified where precise d.p. worms are frequently required. The last value, 22/7 is extensively used; but many values in Table V are based on the values (5), (6), and (7) above, and involve less error than the value 22/7, without the necessity of special gears.

The gear train for d.p. 10 on a lathe with a lead screw $\frac{1}{2}$ in. for instance, is obtained by using the value (5) above:

$$\frac{\pi}{10} : \frac{1}{2} = \frac{\pi}{5} \triangleq \frac{5 \times 49}{6 \times 13 \times 5} = \frac{7 \times 7}{6 \times 13} = \frac{105 \times 35}{90 \times 65}.$$

It will be found similarly that the gear trains for d.p. 13 and d.p. 8 on a lathe with a lead screw $\frac{1}{2}$ in. are derived from the values (6) and (7) for π.

The value (2) leads to several suitable gear trains if a gear with 58 teeth, for example, is provided for; its application is, however, more restricted than that of the value (3).

The gear 97 is not only useful for cutting diametral pitch threads, but also for cutting metric module threads, and it is assumed in Table V that the special gear 97 is available.

The value (4) can be applied in many cases; many lathes are equiped with a gear 127, and Table V assumes that the gear 127 is also available.

Example 26. To cut a single start worm, d.p. 8, on a lathe with $\frac{1}{2}$ in. lead screw.

The corresponding circular pitch (or, to use a better term, "linear pitch") is $\pi/8$ inches, and the required ratio is $\pi/8 \div \frac{1}{2} = \pi/4$.

The value (1) for π leads to the gear train .. .. $\dfrac{5 \times 71}{4 \times 113} = \dfrac{100 \times 71}{80 \times 113}$

The value (2) for π leads to the gear train .. .. $\dfrac{13 \times 29}{30 \times 16} = \dfrac{65 \times 58}{60 \times 80}$

The value (3) for π leads to the gear train .. .. $\dfrac{97 \times 2}{13 \times 19} = \dfrac{97 \times 50}{65 \times 95}$

The value (4) for π leads to the gear train .. .. $\dfrac{21 \times 19}{4 \times 127} = \dfrac{105 \times 95}{100 \times 127}$

The value (7) for π leads to the gear train .. .. $\dfrac{95}{11 \times 11} = \dfrac{95 \times 50}{55 \times 110}$

The value (8) for π leads to the gear train .. .. $\dfrac{11}{14} = \dfrac{120 \times 55}{80 \times 105}$

The special gears 97 and 127 in some cases enable better approximations than the normal gears, even if the values (3) and (4) for π cannot be used. The gear train in Table V for d.p. 12 and $\frac{1}{4}$ in. lead screw, where the gear 97 has no connection with the value (3), is almost as exact as gear trains based on that value.

Similarly, the gear trains for d.p. 5 are not based on the value (4).

An interesting combination of gears is to be found for d.p. 1 on a lathe with $\frac{1}{4}$ in. lead screw; both the gears 97 and 127 are used there.

In countries where the metric system is used, worm gears are, as all other gears, based on the metric module. The linear pitch of a worm of module m, is $m\pi$ millimetres or $\dfrac{m\pi}{25{\cdot}4}$ inches. Some approximate values for $\dfrac{\pi}{25{\cdot}4} = 0{\cdot}1236842$ are

(1) $\dfrac{47}{4 \times 95} = 0{\cdot}1236842$; error 0·00 per 1,000.

(2) $\dfrac{10 \times 20}{33 \times 49} = 0{\cdot}1236858$; error 0·01 per 1,000.

(3) $\dfrac{17 \times 11}{24 \times 63} = 0{\cdot}1236772$; error 0·06 per 1,000.

(4) $\dfrac{5 \times 19}{24 \times 32} = 0{\cdot}1236979$; error 0·11 per 1,000.

(5) $\dfrac{11 \times 19}{26 \times 65} = 0{\cdot}1236686$; error 0·131 per 1,000.

(6) $\dfrac{3 \times 127}{44 \times 70} = 0{\cdot}1237013$; error 0·133 per 1,000.

(7) $\dfrac{12}{97} = 0{\cdot}1237113$; error 0·21 per 1,000.

(8) $\dfrac{34}{11 \times 25} = 0{\cdot}1236364$; error 0·39 per 1,000.

(9) $\dfrac{5 \times 22}{7 \times 127} = 0{\cdot}1237345$; error 0·40 per 1,000.

The first value, with practically no error, requires a special gear 47 or 94, which is justified where module threads are frequently required. Where the special gear 97 is available, the value (7) can be applied. In many cases, however, better approximations can be obtained by the application of the values (2), (3), (4), or (5). Where a gear 127 is available, the value (6) in some cases leads to good results.

Here again, Table V indicates the gear trains for the best obtainable approximations. Interesting gear trains are for $m = 1{\cdot}25$ (lead screw $\frac{1}{4}$ in. pitch), and $m = 3{\cdot}25$ (lead screw $\frac{1}{2}$ in. pitch), where both the gear 97 and 127 are used.

(*To be continued.*)

Method for Measuring the Mass Flow of Gases or Liquids in Closed-Duct Systems

By F. C. L. VAN VUGT. (From *De Ingenieur*, Vol. 62, No. 8, February 24, 1950, pp. *O* 5-*O* 12, 7 illustrations.)

Two new types of mass flow meters for the measurement of the mass flow of gases or liquids per unit time have been developed in the Technical Physics Laboratory of the Technical University of Delft. These mass flow meters can also operate under pulsating flow conditions or with media of variable density.

Most flow meters are designed for fluids of constant average density. In cases where a number of measurements has to be taken at short time intervals in order to obtain good results, continuous measurements are preferable, but the measuring apparatus becomes more complicated.

The various types of measuring devices previously available have made use of one of the following four methods:—(1) counting of a number of pre-determined volumes (chamber-type meters); (2) measurement of the average rate of flow (vane-type meters); (3) measurement of the rate of flow at a given position in a representative cross-section of the stream, by creating a stagnation pressure in the path and determining its value (Pitot tubes); (4) measurement of a pressure difference Δp created in the stream by a contraction in the main duct (Venturi-type meters).

Of the four methods mentioned, the last has been extensively applied for rate-of-flow measurements in physical and chemical processes. Method (3) is similar in principle to method (4), as the product density × velocity squared which is measured can be regarded as the equivalent of a pressure difference Δp. These methods give results which are not reliable when the density of the fluid is not accurately known, or when large-amplitude periodic velocity fluctuations have to be taken into account.

An ideal solution to the problem would be a measuring instrument capable of measuring the specific momentum ρv_L of the flow, for both gases and liquids. The product ρv_L also represents the average density of the mass flow per unit time, expressed in kg-mass/(m² × sec) or lb-mass/(in² × sec). The rate of mass flow could then be obtained by multiplying the result by the cross-sectional area A_L of the liquid passing through the duct.

This principle has been applied in the design of the ρv_L meter or mass flow meter, described hereunder.

BASIC THEORY OF THE MASS FLOW METER

The new mass flow meter has a measuring element built in the main duct system. It uses a differential pressure gauge to measure the pressure difference between two branches of the flow, into which the main stream is divided. It will be shown in the following lines that the measured pressure difference is equivalent to the mass flow density ρv_L, if the simplifying assumptions of Bernoulli's theorem are valid. The flow through the various cross-sections of the ducts will then be in accordance with the relation

$$p + \frac{1}{2}\rho v^2{}_L = \text{constant},$$

where p = static pressure, ρ = mass density of the medium, and v_L = velocity of flow through a cross-section of the duct.

In the measuring element of the mass flow meter, the main stream coming from the duct system is symmetrically divided over a short distance into two equal branches passing round the periphery of a disc or cylinder, the two branches being united again into a single stream behind the disc. As a result of the uniform velocity of rotation of the disc, a circulating stream of constant velocity is superimposed on the two partial flows, acting against the flow in one branch and assisting the flow in the other.

If v_c = velocity of circulation [m/sec], v_d = velocity of either of the partial streams, and v_+ and v_- = resultant velocities, then

$$v_+ = v_d + v_c \text{ and } v_- = v_d - v_c \quad .. \quad (1)$$

Centrifugal effects can be neglected, provided the channels are given a relatively large radius of curvature, and we can then write for the flow through two cross-sections symmetrically situated with respect to the main axis of the flow:

$$p_+ + \frac{1}{2}\rho v_+{}^2 = p_- + \frac{1}{2}\rho v_-{}^2 \quad .. \quad (2)$$

where p_+ and p_- are the static pressures in the two cross-sections. Substituting the values for v_+ and v_- in eq. (2), we obtain

$$p_- - p_+ = 2 v_c \rho v_d = C_1 \rho v_d = C_2 \rho v_L \quad [\text{Newton/m}^2]^* \; .. \quad (3)$$

The mass flow density ρv_L is thus actually determined as the equivalent of a pressure difference $p_- - p_+$ divided by a constant. C_1 and C_2 are instrument constants. Thus, the rate of mass flow q is given by

$$q = A_L \rho v_L = C_3 (p_- - p_+) \quad [\text{kg-mass/sec}] \quad (4)$$

This shows that with this method reliable measurements can be obtained even in cases of pulsating flow for media having a constant or variable density, as in all cases the average value is

$$\overline{q} = A_L \overline{\rho v_L} = C_3 (\overline{p_- - p_+}) \quad [\text{kg-mass/sec}] \; .. \quad (5)$$

DESCRIPTION OF TWO PROTOTYPES

General Features.—The new mass flow meters are so designed that the two channels are mirror images of each other, and the circulating stream has a constant velocity throughout the measurement. The arrangement is based on the use of a rotating surface in the main stream, and this surface entrains the gas or liquid in its uniform rotary motion and thus creates a circulating stream of constant intensity.

The meter casing is preferably of rectangular cross-section and is designed with a central recess in which the rotor is situated. The rotor is a comparatively long cylinder fitted with very little clearance between its end surfaces and the casing. The rotor shaft is arranged perpendicularly to the direction of the main stream and is driven by an electric motor.

A gradual transition from the circular cross-section of the main duct to the rectangular section of the flow-meter casing has been provided, as this gives a more favourable velocity distribution at the inlet and outlet ends of the meter, and minimises pressure losses caused by eddies. The apparatus thus designed will measure the difference between the two static pressures p_+ and p_- occurring in the channels, in two corresponding cross-sections perpendicular to the direction of the main stream.

The profile of the two symmetrical channels around the rotor does not vary in the direction of the rotor axis, but their height, in the direction of flow, can vary according to particular requirements.

* 1 Newton = 1 Joule/m ≃ 0·102 kg-weight.

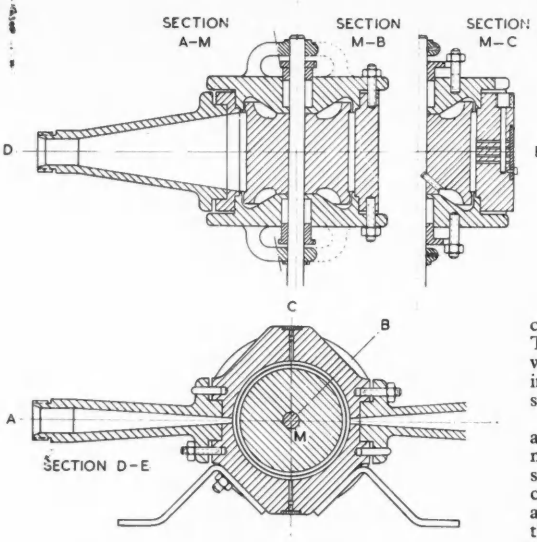


Fig. 1. **Details of mass flow meter No. 1.**

Prototype No. 1. This first design, which is shown in Fig. 1, was developed for the measurement of fluids flowing under low-pressure conditions. The measuring channels have a constant height and the meter is provided with two tapered transition sections so that the main stream can spread out over the full length of the rotor cylinder.

The sides of the casing are designed as removable covers, and their outer annular surfaces also constitute one half of the lateral end-surfaces of the measuring channels. The other half of these surfaces is formed by the protruding edges of the rotor periphery as shown in Fig. 1. Flow leakages have been avoided as far as possible by using very small radial clearances at the edges of the covers.

The two pressure-measuring passages are symmetrically arranged and situated half way up on the sides of the channels, where the compression effects due to the two covers are reduced to their minimum values.

The prototype No. 1 was subjected to extensive tests in order to determine whether the basic principle thus applied would yield reliable results and to study the influences which might have a disturbing effect on the linear relationship between the mass flow velocity and the measured pressure difference. Fig. 2 shows some typical results obtained with oil having a viscosity of 15 centistokes as the flowing medium. The solid lines represent measured results, whereas the dotted lines give the theoretical values calculated for a constant angular velocity of the rotor. The results clearly indicate the existence of disturbing influences. Fig. 2 also indicates the range over which sufficiently accurate results can be obtained, and it also shows that accuracy increases with rotor speed and decreases with increasing flow velocities.

The flow resistance of this prototype was found to be more than twice as great as the corresponding pressure difference between the two measuring channels. An investigation of the flow conditions at the inlet and outlet ends of the meter, and of the pressure distribution on the rotor periphery, showed that these high resistances were produced by (1) eddies formed at the two stagnation points at the channel ends, and (2) pressure losses in the measuring channels.

Both these results were taken into account in developing the second prototype, and the solution adopted was to design the transition duct portions with increased cross-sections at the stagnation points at the two ends of the meter.

Prototype No. 2. The working pressure was limited to a maximum of 15 bars (approximately 15 atmospheres). Design details are shown in Fig. 3. In this prototype the measuring channels are of constant width and varying height. Various experiments showed that maximum measured differential pressures are obtainable when the ratio of maximum to minimum channel height is between 4 and 12. For this design the value chosen is 4·5.

The two carrying blocks are built integral with the covers. This has the advantage of improving the centering of the rotor in the casing, and hence of reducing the measurement error. The medium being measured is used as a lubricant, with this arrangement, but this is usually no drawback in view of the very low bearing pressures of the rotor shaft.

The sealing rings of the rotor and the covers are arranged radially instead of axially. This leads to a much simpler construction, with the rotor acting as a self-centering element in the axial direction. Minimum clearance is provided between the rotor end surfaces and the covers, in order to limit the flow leakage along the rotor to a maximum of 1 per cent. As a result of the high working pressures of the flow, the covers have

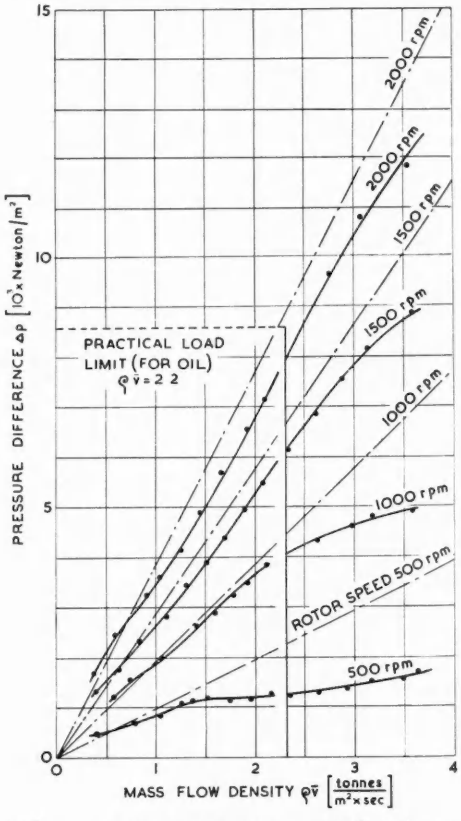


Fig. 2. **Test results obtained with prototype No. 1, for rotor speeds varying between 500 and 2000 rpm.**

been made relatively thick to reduce cover distortion to a minimum value.

To simplify the design, no separate inlet and outlet transition sections have been provided. The meter is mounted with the electric motor required for the rotor drive on a common base plate.

Test Results. Some of the results of the tests with this second prototype are shown graphically in Fig. 4. The tests were carried out under various load and speed conditions, and with various rotor diameters. The graphs clearly show the linear relationship required between the measured differential pressure and the mass flow intensity. Furthermore, the pressure losses are half of those of prototype No. 1. The test results can, therefore, be regarded as very favourable.

Further tests were made to determine the behaviour of the meter under pulsating flow conditions. For this purpose, a controllable rotating vane was placed in the duct system ahead of the flowmeter; in this way it was possible to produce large-amplitude pulsations in the flow. The results showed that there were no appreciable differences in the measurement characteristics for either steady or pulsating flow.

THEORETICAL CONSIDERATIONS

Although the flow equations previously obtained have been based on simplifying assumptions which represent only approximately the conditions in the flow meter, when these equations are used to determine the pressure distribution around the rotor periphery the calculated values obtained are in good agreement with the measured results.

A brief analysis will now be given of some particular features of these mass flow meters.

The calculations previously made have shown that the differential pressure measured in the mass flow meter is related to the mass flow intensity by the equation

$$\Delta p = 2 v_r \rho v_d \text{ [Newton/m}^2\text{]} \quad .. \quad .. \quad (3)$$

For both prototypes, the velocity v_r of the circulating

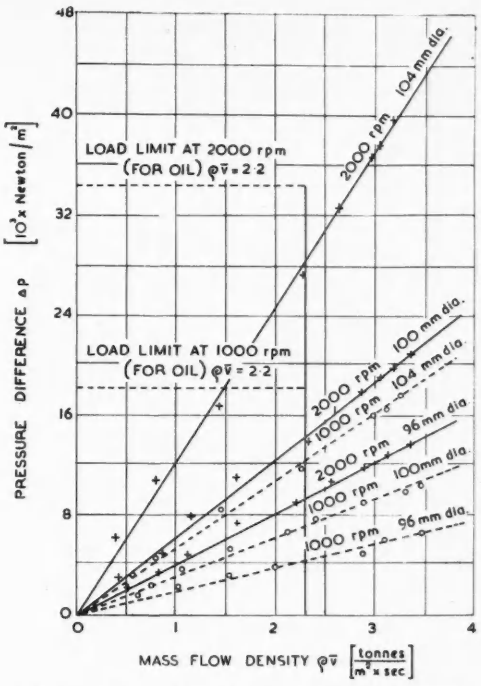


Fig. 4. **Test results obtained with prototype No. 2, for rotor speeds of 1000 and 2000 rpm, with rotors of various diameters.**

stream can be assumed to be proportional to the angular velocity ω of the rotor, so that

$$v_r = a\omega \quad \text{[m/sec]} \quad .. \quad .. \quad (6)$$

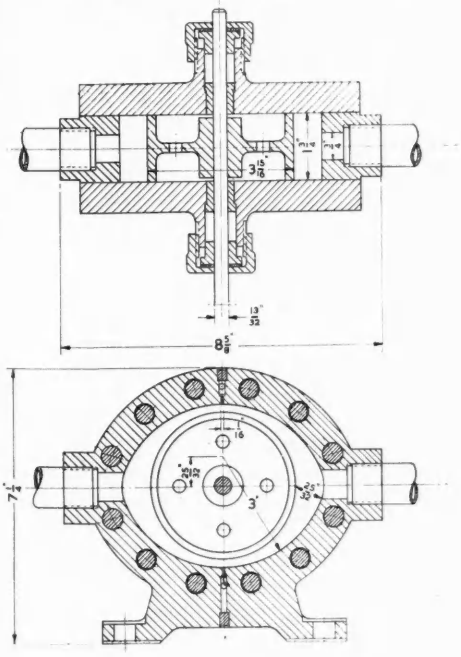
where a can be regarded as a constant. For the mass flow intensity we then obtain

$$q = \frac{C'}{\omega} \Delta p \quad \text{[kg sec]} \quad .. \quad .. \quad (7)$$

where C' is the flow meter constant.

This equation shows that accurate measurements require a constant angular velocity of the rotor, and the measuring range of the meter will depend on the angular velocity and/or on the flow meter constant.

The actual pressure distribution along the rotor periphery depends on three effects to which a fluid element in a measuring channel is subjected: (1) an entraining effect exerted by the rotor; (2) a ram effect caused by the stationary side walls of the channels; and



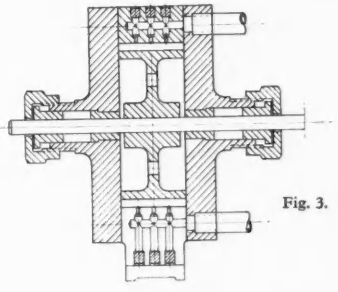


Fig. 3. **Details of mass flow meter No. 2.**

(3) a static pressure effect due to the pressure drop in the direction of flow.

The solution of the differential equation for the pressure distribution based on these three effects leads to an elliptic integral when using the exact relation for the variation of channel height for the practical case where the channel profile is determined by two circular arcs. To avoid this complication, an approximate equation can be used for the variable channel height :—

$$d = \frac{1}{A - B \cos 2\varphi} \quad .. \quad .. \quad (8)$$

where d = channel height [m], A and B being parameters and φ the central angle of the figure. When $\varphi = 90°$, the channel reaches its maximum height $d_0 = 1/(A + B)$

As the rotor radius and the minimum value of d are known, the channel profile is then determined by the choice of one value for A or B. It is thus possible to obtain an approximate curve which represents with sufficient accuracy the true contour variation of the channel height in the region between 50 and 130 degrees, which is without appreciable effect on the measurements.

An analysis has been made of the pressure variation along the rotor periphery for channels of variable contour, taking account of the three effects previously mentioned. The calculations led to the following conclusions :—

(a) By suitable choice of the value of A or B, for otherwise similar values of ω, ρ, v_L, d_{min}, rotor width and rotor diameter, with channels of variable height a measured pressure difference can be obtained which will be approximately 36 per cent higher than that obtained with a mass flow meter having channels of constant height.

(b) The intensity of the circulating stream varies with increasing values of the mass flow intensity q [kg/sec], without altering the angular velocity of the rotor.

The rate of flow of the circulating stream depends on the flow conditions in the equivalent rectified channel. These conditions are determined by various factors and particularly by the direction of the frictional forces occurring between the rotor periphery and the flowing medium.

At low q-values, the rotor has an entraining effect on the fluid over the entire length of the measuring channel. When the load reaches a certain value q_0, this effect is converted into a ram effect at the position where the velocity of the fluid has its maximum value, that is, in the smallest cross-sections which are connected to the pressure-measuring gauge. The ram effect then extends gradually on both sides, until at a certain value q_1 the direction of the frictional forces in the measuring channel is reversed. This is the so-called " primary measurement range ": it begins at zero load ($q = 0$) and ends at the value q_1, for which reversal of the frictional forces occurs at $\varphi = 90$ degrees.

Within this range the peripheral velocity of the rotor is everywhere greater than, or equal to, the velocity of the fluid in the rectified channel. The circulation intensity remains practically constant over this range, and its value (per unit length of the rotor) is given by

$$C = \frac{A\, v_0}{2A^2 + B^2} \quad [m^2/sec] \quad .. \quad (9)$$

where v_0 = peripheral velocity of rotor. The mass flow intensity in this range is determined by

$$q = \frac{(2A^2 + B^2)\, C''}{A\,(A + B)\,\omega} \Delta p \quad [kg/sec] \quad (10)$$

where C'' is the new flow meter constant.

In the second or intermediate range, the reversal of the frictional forces is completed over the entire length of the equivalent rectified channel. The change takes place over an extremely short range of q-values. In most of this range the intensity of the circulating stream varies with the mass flow intensity q too heavily to be useful for measurement purposes. When the reversal is practically complete the circulation velocity again reaches an approximately constant value, which is maintained throughout the " third measurement range." The circulation velocity in this range is approximately given by

$$C^* = \frac{(2A^2 + B^2)\, v_0}{A\,(4A^2 + 6B^2)} \quad [m^2/sec] \quad .. \quad (11)$$

For mass flow meters with channels of constant height ($B = 0$), the circulation velocity remains constant at all loads, and is equal to $1/(2A)$. For a mass flow meter with varying channel profile and characterised by a value $A + B = 200$ (or $d_{min} = 0{\cdot}005$ metre), the maximum possible decrease of C^* relative to C is 11 per cent, which is obtained when $A = 110$.

(*Concluded on page* 190)

U.S.A.

Engineering Education for Atomic Energy Development

By K. H. KINGDON. (From *General Electric Review*, Vol. 53, No. 2, February, 1950, pp. 7-13).

THIS article deals with the present and possible future needs for technical people in the field of atomic energy. It automatically brings with it some appraisal of the scope of opportunities offered by the development of atomic energy. The comments are restricted entirely to peacetime applications, although the production of fissionable materials such as plutonium is, of course, of fundamental interest for both military and peacetime uses.

There are three major fields of effort in the non-military applications of atomic energy :

(1) the production of raw materials ;
(2) the utilisation of radioactive materials produced by nuclear reactors in science and industry ;
(3) the production of useful power.

The third of these fields will be the one given major consideration in this article, but all of them will be included in the aim to gain an understanding of what kinds of technical people, and how many of them, may be needed for development in the future.

1. PRODUCTION OF RAW MATERIALS

Uranium ore, the primary raw material of atomic energy, is distributed widely throughout the earth's crust, but most of the deposits are of such low grade that they are uneconomically expensive to refine.

The techniques used so far in the preparation of raw materials have been mainly adaptations of those common to the mining and ore refining industries. The ore is dried, crushed, ground, weighed, sampled, packed and chemically analysed. The analysis establishes the uranium content of each batch of ore.

After it is analysed the raw material goes to chemical refineries for the production of uranium hexafluoride gas or uranium metal slabs. By ordinary industrial operations in metal-working plants, the metal slabs are

converted into rods of suitable size and shape for use in reactors.

Although much higher standards of purity are required, the atomic energy raw material industry is not very different from other refining and chemical industries up to the point of use of the material in reactors. It is expected that men with a conventional training of mining, and chemical engineers and metallurgists will be able to handle these jobs. How many such men will be needed in the future depends on the extent of the development of atomic power.

2. UTILISATION OF RADIOACTIVE MATERIALS PRODUCED BY NUCLEAR REACTORS

Radioactive isotopes for scientific, medical, agricultural, and industrial uses are at present the main peacetime contributions of the development of atomic energy. Medicine and biology at present constitute the largest fields for the use of radioisotopes, and absorb about 45 per cent of the isotope shipments. Research in chemistry and physics accounts for another 36 per cent of the shipments, while the uses which are classified as industrial and metallurgical amount to only some 10 per cent. It is to be expected that the industrial uses of radioisotopes will expand as time goes on, but many of the industrial applications which at first sight seemed important have not turned out well on further study.

All applications of radioisotopes require special instruments, such as radiation counters, ionisation chambers, scaling circuits, personnel radiation meters, film badges, lead bricks, and tools for the remote handling of radioactive materials. The demand for these special instruments is, of course, common to all activities of atomic energy work. This demand has led to the building up of a sizable new radioactive instrument business, which employs physicists, chemists, and engineers who have specialised in instrumentation.

The people who are making use of these radioactive materials can hardly be classified as a new group of technical people working mainly in the field of atomic energy. They are primarily workers in the medical, biological, and other sciences, who are making use of by-products of atomic energy as a new tool in their respective fields.

3. THE PRODUCTION OF FISSIONABLE MATERIALS AND OF USEFUL POWER IN NUCLEAR REACTORS

The production of useful power from atomic energy and the creation of new fissionable material for use as fuel are accomplished by means of nuclear reactors. The concept of a nuclear chain reaction, as used in a nuclear reactor or pile, has become a familiar one. A nuclear reactor is a very special assemblage of fissionable fuel, moderating material to control the energy of the neutrons, coolant to remove the heat, various structural materials to form the supporting framework, shielding to confine the intense radiation, and some means for controlling the rate at which the nuclear reaction proceeds.

The consumption of nuclear fuel in the reactor is similar in many ways to the burning of conventional fuels in a furnace, and it is instructive to compare corresponding items in the two cases. This comparison will also offer a convenient way of assessing the technical skills which will be required in the design, construction, and operation of the power and breeding reactors of the future.

(i) *Fuel.* The fissionable material so far used as a fuel has been in the form of a metal or an oxide or a solution of a chemical compound. In addition to the requirement of purity, nuclear fuel in most cases will need a relatively large amount of fabrication to fit it for use in the reactor. This arises mainly from three causes :

(1) the need to control the radioactively dangerous fission products formed when the fuel is consumed.
(2) the extremely high energy concentration in the fuel.
(3) as a corollary to (2), the desire to obtain relatively enormous heat tranfer from a given amount of nuclear fuel as compared with an equal amount of ordinary fuel. It will be evident from this that much more metallurgical and mechanical skill will enter into the fabrication of nuclear fuel than is the case with conventional fuels.

(ii) *Heat Transfer.* In a nuclear reactor, there are two compelling reasons for getting the maximum possible heat transfer out of the smallest possible volume :

(1) It is evident that if the fissionable fuel is compactly arranged, there will be less chance for neutrons to escape from the reactor than if the fuel were rather widely dispersed, and that, therefore, a compact structure will conserve fuel.
(2) Nuclear fuel is so expensive that it is profitable to invest a considerable amount of mechanical and metallurgical effort in getting the minimum possible amount of heat out of a given amount of this fuel.

The existence of these boundary conditions implies that a great amount of engineering effort will be put into the heat transfer problems of nuclear reactors just as considerable effort has gone into the engineering design of a compact radiator for automobiles. The services of skilled heat transfer engineers will be at a premium in the nuclear reactor field for some time.

(iii) *Corrosion.* The materials of the nuclear reactor are subject to conventional corrosion by the gaseous or liquid coolant, but with the added difficulty that maintenance of the inner part of the reactor is extremely difficult, if not impossible, on account of the intense levels of radioactivity which prevail there even after the reactor has been shut down. The solution of this problem will require the services of chemists and metallurgists.

(iv) *Radiation Stability.* The problem of radiation stability has no counterpart in the conventional furnace. The interior of a nuclear reactor is filled with almost inconceivably intense radioactive radiations, particularly gamma rays and neutrons. Both these radiations have the ability to penetrate considerable thicknesses of the materials within the reactor and to cause displacements of the atoms of these materials. These displacements may result in a kind of internal corrosion of the materials in the reactor which would make them unfit for prolonged operation. Much experimentation by physicists and metallurgists will be needed to dedetermine which structural materials can satisfactorily withstand the intense radiation conditions within a reactor.

(v) *Neutron Economy.* Only a strictly limited supply of neutrons is produced by fission, but these neutrons are essential for the maintenance of the chain reaction and for the production of new fissionable material which is so vital to the development of a large atomic power industry. It is evident, therefore, that the neutrons must not be wasted by absorption in materials or poisons which play no vital part in the overall economy of the nuclear reactor. This condition forbids the use of many of the conventional and well-tried materials of engineering in certain types of nuclear reactors, and brings with it the necessity of exploiting new and untried materials. A large amount of metallurgical work will be needed to bring the technology of these new materials to a point where it will be adequate for their extensive use in nuclear reactors. There will also be the need for tests of the nuclear properties of these new materials by physicists.

(vi) *Control.* The concentration of energy in nuclear fuel is so enormous, and the necessary initial charge, or critical mass, of nuclear fuel is so large compared to the rate at which it is burned up, that it probably will seldom be necessary to refuel a power reactor while it is in operation, and it certainly would be difficult to control the rate of energy generation in the reactor by a refuelling process. Instead, some means is used to alter the number of neutrons present in the reactor at any instant, so that this number slowly increases or decreases. The mechanisms by which the control elements are moved involve familiar principles of mechanical and electrical control engineering, and engineers who work effectively in this field may be adequately trained by conventional university courses, followed by practical experience.

(vii) *Introduction of Fuel and Removal of Products.* The problems of feeding and removing fuel are much more complicated for the nuclear reactor on account of the high level of radioactivity which prevails within the reactor even after it has been shut down for some time. This condition requires that the new fuel be introduced, and the spent fuel be removed, by some remotely controlled mechanism. These mechanisms vary greatly in complexity with the type of reactor they are to service. The more complicated ones offer a remarkably fertile field for the talents of the expert mechanical designer.

(viii) *Shielding.* A conventional furnace is boxed in to retain the hot gases and is lined with materials which will not readily transmit heat. This problem of shielding against heat is also present in the nuclear reactor, but, in addition, there is the far more difficult problem of shielding against the escape of the penetrating gamma rays and neutrons. The escaping neutrons must be captured by a nuclear process, and the ability of various substances to capture neutrons is rather well known. The absorption of gamma rays requires materials of high density, so that a gamma ray shield is always heavy. Although it is true that the fundamental principles of shielding are understood, it is also true that the detailed design of a shield for any particular application still requires involved physics calculations and skilled engineering design. This is particularly true if the shield is for a reactor on a ship, where conservation of weight and space is important.

(ix) *Chemical Processing of Used Fuel.* Materials removed from a reactor require elaborate chemical processing. In most cases, some new fissionable material such as plutonium will have been produced. This must be chemically separated and worked into a form suitable for use. Any unconsumed residue of the original charge of the fuel must be recovered, since it is valuable material, and, finally, the fission products must be separated and disposed of. Some of the fission products absorb neutrons, but in an entirely useless way, so that if they are not periodically removed from the reactor they will impair its efficiency. All of these materials coming out of the reactor are highly radioactive, and the chemical operations carried out on them must be remotely controlled. This, then, is a challenging field for the chemist and the chemical engineer, and revolutionary progress must be made before it will be possible to have an economically sound atomic power industry.

(x) *Waste Disposal.* The sea is big enough to accommodate safely all the fission products that would be produced by a sizable atomic power industry, provided that they could be uniformly distributed throughout the oceans of the world. This is difficult to guarantee, and there is always the serious chance that some fish will eat the concentrated fission products or that they will be picked up by some marine vegetable growth. The tolerance of some forms of marine life for radioactivity is much greater than human tolerance, so that procedures cannot be allowed which might permit humans to catch and eat highly radioactive fish or marine flora. In view of such considerations, it seems safer to concentrate the liquid and solid radioactive waste to small volumes and to store them until a national policy for the safe disposal of these wastes is decided upon. A possbile procedure might consist in mixing with concrete and dumping the concrete blocks into the sea.

The problem of waste disposal is thus another one on which a great deal of chemical talent can be used.

(xi) *Production of New Fuel.* The conventional furnace consumes fuel without producing a new supply of fuel to replace that burned up. One of the most remarkable things about a nuclear reactor is that there is the possibility of using some of the excess neutrons produced in fission to make a fresh supply of fissionable material. The new fissionable materials are made by the interaction of neutrons with uranium-238 or with thorium, which are called fertile materials and are specially placed in the reactor for this purpose. If the amount of new fissionable material produced exceeds the amount burned up, the reactor is said to be a breeder. The only useful fissionable material found in nature is uranium-235, and this is rather rare, since it constitutes only one part in 140 of natural uranium. The possibility of breeding opens up the prospect of converting all the uranium-238 and all the thorium into useful fissionable material and thus increasing the supply of fissionable material by a factor of several hundred. Thus, the satisfactory development of breeding is of the greatest importance for the establishment of a large atomic power industry, and the development of power-breeding reactors will be a major task for technical people working in this field.

4. KINDS OF TECHNICAL PEOPLE NEEDED FOR ATOMIC POWER DEVELOPMENT

This brief survey of reactor problems has shown something of the variety of technical people needed for atomic power development. It will be instructive to take a closer look at the staff of the Knolls Power Laboratory. This laboratory is entirely concerned with the development of atomic power and has no research work going on in other fields. The technical personnel of the laboratory is divided among the various functional divisions as shown below :

Administration	3 per cent.
Physics	22 ,, ,,
Chemistry	18 ,, ,,
Chemical Engineering	16 ,, ,,
Metallurgy	17 ,, ,,
General (electrical, mechanical, heat transfer)	21 ,, ,,
Health Physics	3 ,, ,,

Only about 15 per cent (2/3 of the physics groups) of the staff need to have an expert knowledge of reactor neutron physics and general neutron physics. Most of the chemists and chemical engineers need knowledge of radioactive materials in addition to their usual training.

Most of the physicists have had good graduate training in nuclear physics and have acquired the necessary specialised reactor neutron physics in a relatively short time in this laboratory or elsewhere. Most of the chemists were specialised in various standard fields and acquired their knowledge of radioactive materials in this laboratory or in others. It is not possible at present to get much of this type of " hot chemistry " training in universities, since the special facilities for handling large amounts of radioactive materials are available in only a few places.

All the engineers have had the standard kinds of

training in their respective fields. It was thought that some general instruction in the field of atomic energy was desirable for the engineers and certain others of the staff, and two kinds of courses have been given in the laboratory to achieve this end. The first "advanced" kind of course aimed at making certain selected engineers familiar enough with the neutron physics of reactor design to enable them to carry out a large part of the engineering design of a reactor without help from physicists; but in the present state of development it seems easier to employ physicists for much of this work, and this course has been discontinued for the present. The second kind of course is of a more popular nature and is the sort of course through which the majority of the staff should go. The work of this course is about evenly divided between nuclear physics, reactor neutron physics, and reactor technology, including metallurgy.

The development of an atomic power industry is analogous in many ways to the petroleum industry. It will have similar problems of developing and, thus, this comparison, faulty though it is, enables one to get some sort of perspective on the technical manpower

(*Concluded on page* 188)

FRANCE

A New High-Frequency Fatigue Testing Machine

By R. JACQUESSON and P. LAURENT. (From *Revue Générale de Mécanique*, Vol. 34, No. 13, January, 1950, pp. 31-33, 5 illustrations.)

HIGH-FREQUENCY fatigue tests at 5000 to 60,000 cycles per minute become more and more important as the speed of machinery is increased. These tests permit of studying the influence of higher harmonics of vibration; their high speed may prevent slip in the material, so that test pieces either fail rapidly at points of flaws or indefinitely sustain higher loads than they would sustain at low speeds. Due to difficulties in design, high-speed fatigue testing has, up to now, usually been limited to push-pull or rotary bending tests. The present article, however, describes a new machine designed for repeated, fluctuating, and alternating, torsion and bending fatigue at high speeds. The amplitude of vibration can be adjusted during tests, so that the effects of a temporary overloading or of rest periods may be studied.

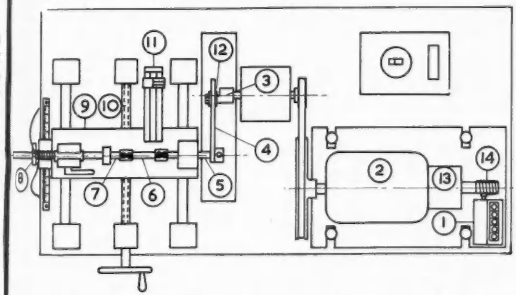


Fig. 1. **Schematic arrangement of testing machine.**
(1) Revolution counter; (2) motor; (3) constant-speed shaft; (4) slotted lever; (5) oscillating shaft; (6) specimen; (7) stationary chuck capable of sliding axially; (8) current interruptor; (9) movable platform; (10) spindle for platform adjustment; (11) fixture for bending-test specimens; (12) ball bearing of eccentric; (13) speed control; (14) worm gear.

Fig. 1 illustrates the schematic arrangement of the testing machine. The motor (2) supplies $^1/_2$ hp at speeds adjustable between 1000 and 6000 rpm. A Vee-belt transmits the movement to a constant-speed shaft (3) carrying an eccentric of constant throw at its free end. The belt pulley of shaft (3) is interchangeable, so that this shaft may run at 3 or 6 times the motor speed. The belt is tensioned by adjustment of the motor base-plate; slip was found to be entirely negligible. The eccentric is fitted with ball bearings sliding in the slot of a lever (4) made of a high-strength light-alloy and pivoted on the oscillating shaft (5). This part of the machine is enclosed in a gear box and gravity lubrication is provided. The angle of oscillation of shaft (5) thus depends on the distance between shafts (3) and (5), and this can be changed, during running, by adjustment of the position of the whole test platform by means of the handwheel-operated spindle (10). The maximum obtainable angle of oscillation is, of necessity, very small for high-speed tests, and does not exceed $\pm$ 15 degrees, so that short test specimens are to be used if high stresses are required. Shaft (5) carries the vibrating chuck for the specimen (6); the stationary chuck (7) of the torsion specimen can slide axially, and a weak spring withdraws this chuck when the specimen breaks, thus interrupting the motor current. For bending tests, the specimen is arranged in a position normal to the axis of oscillation and clamped in the stationary chuck (11) at an adjustable distance from the axis of the oscillating shaft (5). Chuck (11) is electrically insulated and the motor control current passes through the specimen, so that the motor is again stopped when the specimen breaks. Bending fatigue and standing waves can thus be studied in wires or thin metal strips. In order to eliminate vibrations of the main frame of the machine, shaft (3) with the eccentric has been balanced very carefully, and the motor with its revolution counter as well as the entire frame are carried on rubber shock-absorbers. In actual practice, the machine has been running continuously at a speed of 18,000 oscillations per min.

The machine, although small in overall dimensions, is easily adaptable to the most varied uses. The fact that the amplitude of oscillation as well as the length of the specimen can be adjusted is a great advantage compared with other testing machines. Torsion fatigue tests can also be taken with a superimposed constant tension. It must finally be stated that test results remain comparable only if play in bearings is prevented. The slotted lever, which is simple to manufacture, but wears more rapidly than the other parts, should therefore be replaced after about every 100×10^6 oscillations.

Examples are given of some investigations carried out with this machine. The effect of the speed of oscillation on the torsion fatigue strength of aluminium was investigated on wires of 1 mm diameter and 6 mm length. The influence of the length of a specimen on the torsion fatigue limit was investigated under constant maximum stresses and was found to be very small. Long test-pieces tend to fracture more quickly, as the number of flaws they may contain is larger. An interesting study of torsion tests on electrolytically polished sheet strips of aluminium, 10 mm in width by 1 mm in thickness, permitted a microscopic examination of the development of fatigue cracks during the tests. A pronounced difference between the fatigue behaviours of a cold-worked material and an annealed material could thus be shown. The orientation of the test axis to the direction of rolling was proved to be very important.

DESIGN TRENDS AND TH

OF MACHINE TOOLS

MILLING AND GRINDING MACHINES PART VI

(b) *Centreless Internal Grinding*

Similar to the cylindrical centreless grinding, and having the same production advantages, the operation of centreless internal grinders requires that grinding of the outer diameter be performed before internal grinding, as the roundness of the internal grind depends on the roundness of the outer diameter. The outer diameter rotates between rolls and is driven by a pressure roll (Fig. 150).

Machines of this type are generally fully automatic, since work done on them lends itself well to handling by loading devices.

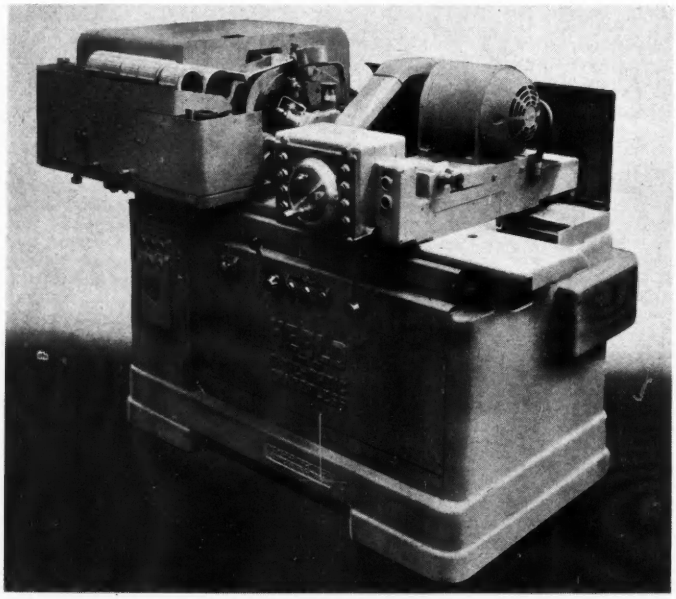


Fig. 151. **Model 281 Heald centreless internal grinding machine.**
Max. outside dia. of work 6 in., max. length of hole that can be ground 4 in.

Such a machine is the American Heald, Model 281, shown in Fig. 151, which is automatic in every element of the grinding cycle, including loading and unloading. The grinder is furnished with " Gage-Matic " sizing, but may also be operated with automatic sizing. Since the cycle is automatic, one operative can handle several machines. The elements of cross-feed cycle and table speeds are separately set for feed rates and amounts by individual graduated knobs. Table stroke is set by self-locking reverse dogs ; the roll mechanism can be set for straight or taper work. All controls are easily accessible and are arranged for the operator's convenience. Knobs and levers are well laid out and marked to indicate their function. The hydraulic table speeds and independent wheelhead cross-slide feeds have graduated dials. The four electric motors have the following specification : main drive 2 hp, 1800 rpm, wheelhead motor 5 hp, 1800 rpm, regulating roll motor with speed reducer $\frac{1}{3}$ hp, 1800 rpm, coolant pump motor $\frac{1}{4}$ hp, 1800 rpm.

SURFACE GRINDING MACHINES

SURFACE grinding machines can be divided into two classes, those having horizontal spindles and those with vertical spindles. The former make use of the periphery of the grinding wheel, the latter use the face of the cup-type wheel. As the contact length between the wheel and the work is generally considerable, wheels are operated at relatively low speeds.

The workholding device on the surface grinder is generally a magnetic chuck for ferrous materials, fixtures being used for non-ferrous materials. Worktables are of either the reciprocating or rotary type. The latter is associated with the vertical surface grinder.

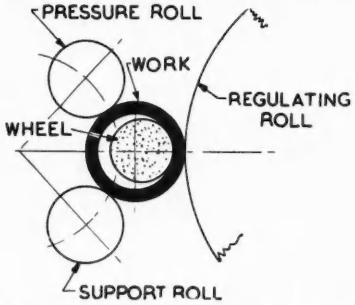


Fig. 150. **Principle of operation in centreless internal grinding.**

THE STYLE

TIBOR HAAS, A.M.I.Mech.E., A.F.R.Ae.S.

Fig. 152 shows a horizontal surface grinder made by the Churchill Machine Tool Company. The worktable has a variable hydraulic traverse and a sensitive hand traverse. An automatic cross-feed with safety trip is provided; cross-feed to work and down-feed to wheel have coarse and fine movements. The wheelhead is detachable and moves on vertical slides. A magnetic chuck can be seen fitted to the table. The chuck can be of either the electro-magnetic or permanent magnet type. This machine is particularly suited for toolroom work, such as die and gauge grinding. It is

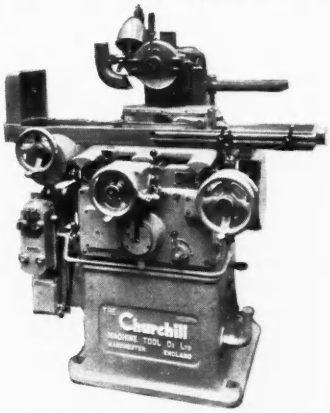


Fig. 152. **Churchill horizontal spindle surface grinding machine, Model NB, 18 in. × 6 in.**

equally useful under production conditions. The Churchill VXA vertical spindle surface grinding machine (Fig. 153) is of robust construction, suitable for automotive work, railway shops, machine tool makers, etc. Where a high degree of accuracy and flatness is required, grinding generally offers the best method of finishing castings and forgings, particularly from the "black." Apart from the better finish obtained, the saving of cost by grinding compared with other machining methods is considerable. Outstanding features of the Model VXA include powerful "built-in" motor drive to the grinding wheel spindle, variable hydraulic table speeds with "run-out" to loading position, permanently protected tableways with automatic lubrication, sensitive vertical feed and quick hand

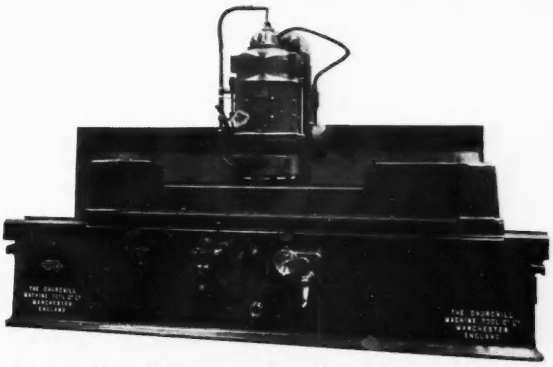


Fig. 153. **Churchill vertical spindle surface grinding machine, Model VXA.**

motion to grinding wheelhead, and efficient wheel and water guards. The segmental chuck with which these machines are fitted has the segments mounted on the periphery. They can be worn down to one-sixth of their original size. Wheel segments are used both for ease of wheel manufacture and to reduce wheel contact with the work. The power available in the vertical spindle wheelhead enables full use to be made of modern abrasive segments. The segmental chuck is extremely rigid, preventing deflection under the heaviest cuts, and can take full advantage of the maximum power. These machines can be supplied with a magnetic chuck, which provides an efficient means of holding single heavy components as well as quantities of small pieces.

Another British vertical spindle machine with a reciprocating table, the Lumsden 70 LCOD, is shown in Fig. 154. Its counterpart, the Lumsden 90LC vertical spindle machine, Fig. 155, is of the rotary table variety. These new machines, developed over a period of forty years, are of exceptionally robust construction and incorporate triplex chain drive to the main spindle,

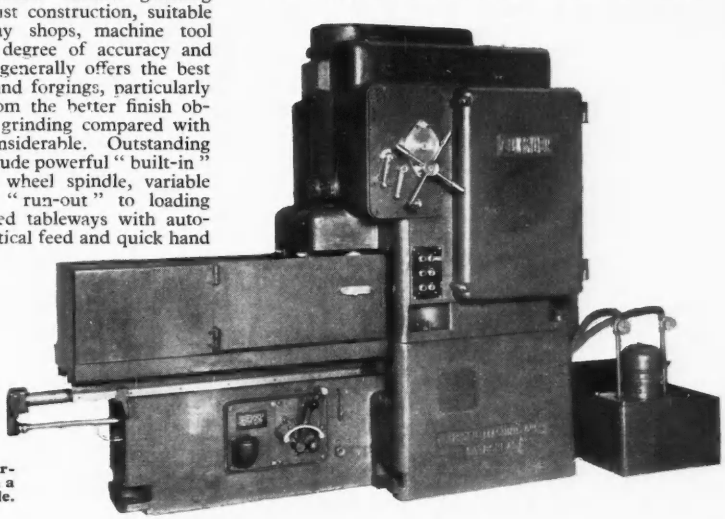


Fig. 154. **Lumsden 70 LCOD vertical spindle surface grinder with a 48 in. × 12 in. reciprocating table.**

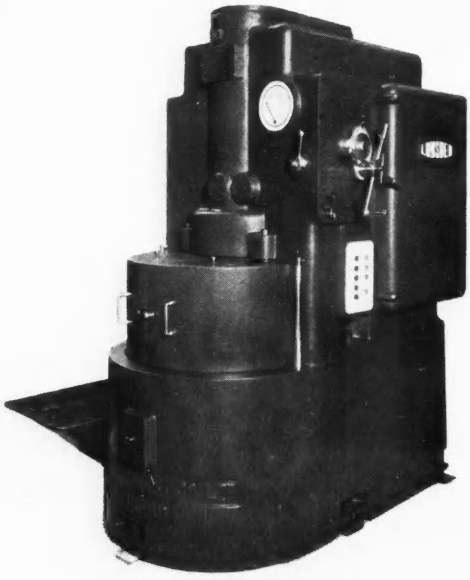


Fig. 155 (above). **Lumsden 90 LC vertical spindle surface grinder with a 24 in. diameter rotary table and a vertical capacity of 12 in.**

Fig. 156 (right). **No. 18 Blanchard vertical spindle surface grinder with rotary table.**

adjustable column, push-button control, improved water service with enclosed coolant tank, and pump lubrication to the tableways.

The Lumsden features of central and enclosed cylindrical ram housing for the heavy flanged spindle and the simple, widely variable automatic feed have been retained, but redesigned on more pleasing lines.

In point of fact, these machines represent best British practice. They may serve to show that there is no need for applied decoration ; good shape and surface can dispense with such visual deception, once function is properly interpreted as an element of appearance.

The No. 18 Blanchard grinder, Fig. 156, has an 18 in. dia. wheel, 30 or 36 in. dia. magnetic chuck, and 25 or 35 hp motor on the spindle. It has power traverse for moving the chuck between loading and grinding positions. Most work is held directly by magnetism. Non-ferrous work can be held by steel blocks which in turn are held magnetically. Irregular shapes are held in magnetic or mechanical fixtures. High power, rigidity, ease of control and good appearance design are features of this machine.

Similar to the Blanchard is the American Mattison No. 36, Fig. 157. It consists of a base, a horizontal slide carrying the magnetic chuck, and a column assembly which includes the grinding wheelhead, head slide, wheel dresser and control panels. The power-driven magnetic chuck is loaded with the work and, while revolving, is traversed into grinding position under the wheel by pressing a push-button. The grinding wheel is fed down onto the work either by hand or automatically, as desired. A predetermined stop or automatic throw-out halts the feed when the finished size of the work is reached. A sealed reflector-type floodlight mounted on the head assembly ensures adequate illumination of the work.

In the Mattison No. 72 surface grinder, Fig. 158, three or four vertical grinding heads are mounted on a central column. A ring-type rotating table carries the work under the

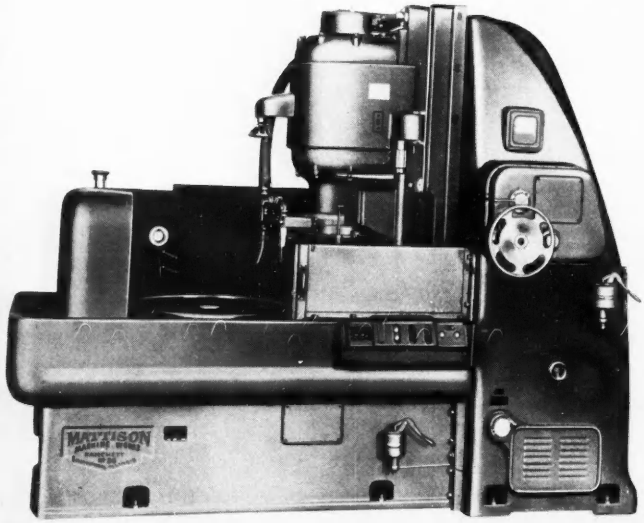


Fig. 157. **Mattison No. 36 vertical spindle rotary type surface grinder, 30 in. dia. chuck, 18 in. dia. wheel.**

Fig. 158. **Mattison No. 72 vertical spindle rotary type surface grinder, 72 in. dia. worktable.**

wheels so that rough, semi-finish, and finish cuts can be taken in one pass through the machine. Automatic or manual fixtures, rectangular or rotary magnetic chucks together with loading and unloading devices enable one operator to turn out more parts than would three general-purpose surface grinders. The machine is equipped with automatic work sizers and wheel feeds and built-in wheel dressers. Controls are centralised and of the push-button type.

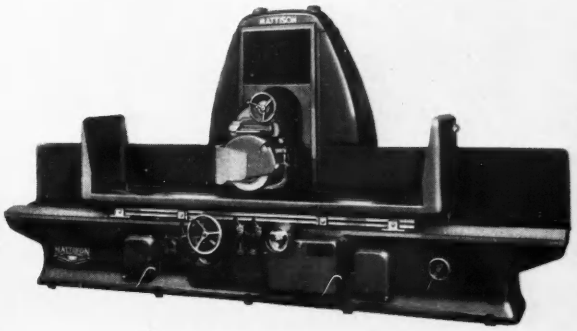
Another Mattison, the plano-type with vertical spindle, Fig. 159, is suitable for grinding surfaces of unusually large areas or lengths to an accuracy of 0·005 in. The table is driven by hydraulic power, which is adjusted for constant feed speeds up to 60 fpm. Bed, table, housing and cross-rail are of heavier and more robust construction than is usual in ordinary planers of equal capacity, a necessary safeguard against inaccuracy under the heavy sustained feeds demanded by this type of operation.

A horizontal spindle plane made by the Mattison Machine Works supplied in finished table top length of 220 in. is among the largest of its type ever built. The largest twin head plano-type of horizontal surface grinder so far built has already been described in this journal[27]. This machine was made by the American Thompson Grinder Company, Springfield, Ohio. Fig. 160 shows a horizontal spindle surface grinder made by Mattison. A characteristic of this grinder is the double-column support of the wheelhead. The sturdy column castings are rigidly bolted and dowelled to the rear part of the base casting, and are reinforced at the top

Fig. 159. **Mattison vertical spindle plano-type surface grinder.**
Table lengths from 72 in. to 220 in., table width from 30 in. to 50 in.

by a connecting brace. These columns, with bearing surfaces on the inside face and dovetail bearing surfaces on the front and back of each column, provide an eight-surface support for the wheelhead. The base is cast as a single unit and is well braced to provide the rigidity so necessary for grinders working to close limits. Feeds are hydraulic and wheel dressing is also done hydraulically.

Fig. 160. **Mattison horizontal spindle surface grinder.**
Table sizes : Width 16 in. to 36 in., length 72 in. to 192 in.



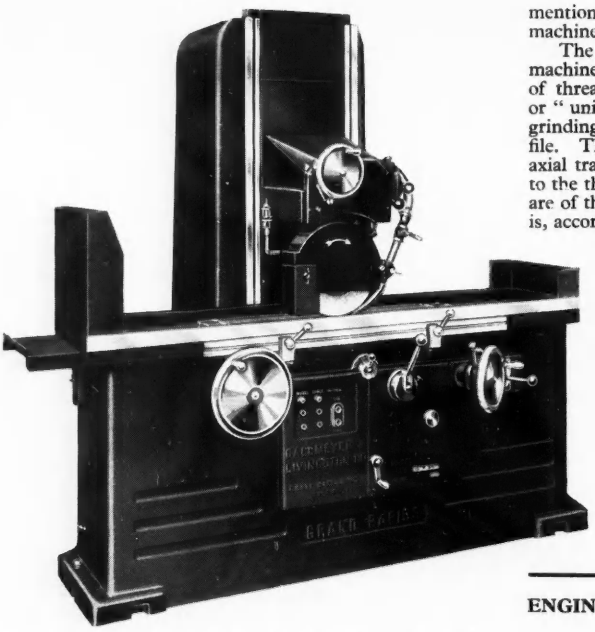


Fig. 161. **Model "A" Grand Rapids horizontal surface grinder.**
Table size 12 in. × 36 in. (Table splash guards removed.)

Another hydraulic feed surface grinder is shown in Fig. 161. It is the American "Grand Rapids" manufactured by the Gallmeyer & Livingston Co. The single upright column which carries the horizontal wheelhead is a one-piece casting. So is the base, which is more than twice the length of the working surface of the table, so that the working surface of the table never overhangs the base in normal operation.

SPECIAL GRINDING MACHINES

Under this heading we shall consider thread grinding machines, profile grinding machines, gear grinders and tool and cutter grinders, whereas, except for a brief mention, no space can be devoted to special-purpose machines.

The thread grinding machines described here are machines designed and used *exclusively* for the grinding of threads. They are either "external," "internal," or "universal," according to the work produced. The grinding wheel is shaped to conform to the thread profile. The work piece is rotated and there is a relative axial traverse between it and the wheelhead amounting to the thread pitch per revolution of the work. Wheels are of the single or multi-rib type and the axial traverse is, accordingly, a full or partial one.

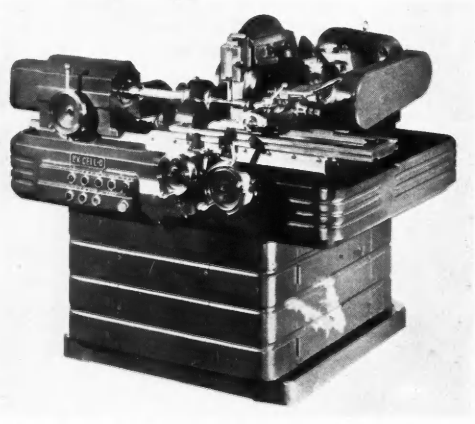


Fig. 162. **Ex-Cell-O Style 31 external thread grinder.**
Max. work dia. 5 in., max. length between centres 18 in.

Fig. 162 shows the American Ex-Cell-O Style 31 external thread grinder suitable for one or two-pass production grinding with automatic stopping or continuous grinding with manual stopping. The front of the grinding wheel is dressed manually for the thread form normal to the work. Arrangements can be made for dressing normal to the helix of the thread.

(*To be concluded next month*).

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ENGINEERING EDUCATION FOR ATOMIC ENERGY DEVELOPMENT

Concluded from page 183.

needs of this possible new industry in relation to the national picture. It is to be expected then that in the United States a considerable atomic power industry would use a few thousand engineers and scientists. By way of comparison it may be noted that the General Electric Company employs in its various laboratories, design, application, and manufacturing divisions, about 9000 scientists and engineers, and that there is a national pool of some 300,000 engineers and 150,000 scientists of all sorts in the United States.

To summarise, it may be stated that most of the technical people to be used in the atomic power effort in the future will need training in special fields of engineering, and in physical, chemical, and metallurgical skill. Perhaps 10 per cent will need the new fission and neutron knowledge of modern nuclear physics. Some of this they will be able to get in universities, but security restrictions and the probably continued general unavailability of nuclear reactors and other expensive and restricted equipment and materials will mean that much of the specialised technical knowledge will have to be obtained on the job. A considerably larger group than the 10 per cent mentioned, and consisting of chemists, chemical engineers, and health physicists, will need practical knowledge of how to handle radioactive materials in bulk. Here, again, this knowledge will probably have to be obtained on the job.

A final word should be said about the calibre of these men. The work they will be doing in the immediate future will be concerned with radically new developments and there will be practically nothing of the handbook type of engineering. The laboratories and organisations responsible for these developments should recruit the highest calibre of engineering talent. Many of the men should be young, with a fresh viewpoint and enthusiasm characteristic of youth. These should be leavened with a sprinkling of more experienced people to supply a background of mature engineering judgment. With a competent staff of this sort and with substantial financial backing, satisfactory progress should be made during the next few years.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

GAS TURBINES

A French Gas Turbine for Automobiles

By C. FAROUX. (From *La Vie Automobile, France*, Vol. 45, No. 1415, March 10-25, 1950, pp. 49-52, 5 illustrations.)

THE "Oredon" gas turbine, built by the Turbomeca Co., of Bordes, has successfully completed two consecutive trials of 150 hours. It develops 150 bhp for continuous running, with a maximum of 170 bhp. The rotor speed is 36,000 rpm, which is reduced by gearing to 6,000 rpm at the output shaft. The total weight is 178 lb. The fuel is kerosene, but petrol is used for starting, about 1 pint being required for each start.

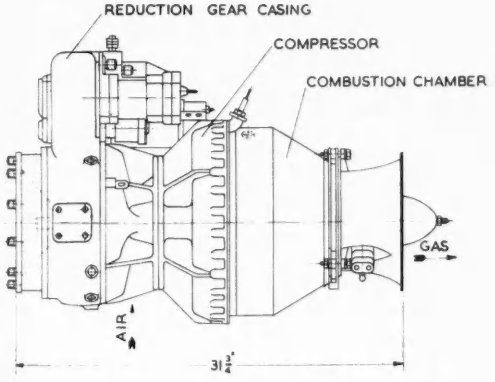

Fig. 1

The compressor is of the centrifugal type, followed by an annular combustion chamber and a two-stage turbine, using a special Turbomeca blade profile. The layout is shown in Fig. 1. The rotor shaft is carried on a thrust bearing at the forward end and a roller bearing at the rear. There is a rotary fuel injection system, supplied through the forward end of the shaft. Cooling for the rear bearing is provided from the primary air (which has by-passed the combustion chamber). Part of this air stream passes down the front face of the turbine disc, through the bearing, and into the jet pipe. After diffusion, the final velocity of the exhaust stream is 300 fps.

METALLURGICAL ENGINEERING

Critical Deformability of Superheater Tubes Made of Cr-Mo-Si-Steels.

By W. MIELENTZ. (From *Brennstoff-Wärme-Kraft, Germany*, Vol. 2, No. 1, January, 1950, pp. 9-13, 20 illustrations.)

SUPERHEATER tubes made of Cr-Mo-Si steels have, in some cases, shown typical cracks after a short time in service, or even during hydraulic pressure tests. These cracks invariably occurred at pipe bends and ran in an axial direction, usually in the neutral zone of bending, but sometimes also in the extreme fibres. The cracks in the neutral zone were found to have started at the bore, and the cracks in the extreme fibre started at the outside of the tube. The material always failed in a brittle fashion, without any signs of previous plastic deformation. A typical analysis of the steels used is: 0·12 C, 0·47 Mn, 1·14 Si, 2·60 Cr, 0·61 Mo, 0·06 Ti, and typical mechanical properties are: Yield stress above 20 tsi, U.T.S. 30 to 40 tsi, elongation 17 per cent on 10 diameters.

Since service conditions cannot have caused these failures, they must have been due to workshop treatment, more specifically to bending over unsuitable bending rolls. Starting from brittle cracks in an axial direction, these fractures cannot have originated from the usual longitudinal bending stresses, but can only have originated from transverse stresses as occur when a circular pipe cross-section is deformed into an elliptical cross-section during bending. This type of deformation leads to a sharper curvature at the neutral zone of bending, and hence to compression stresses at the pipe bore near the neutral zone. Cracks, however, start from points of maximum tensile stress, and it must be assumed, therefore, that the cracks originate when a bent tube is partially or fully straightened, or a deformed elliptical cross-section re-shaped into a circular one. In fact, the elastic spring-back of the pipe, or the straightening effect of hydraulic pressure, may be severe enough to initiate a crack or prepare for its initiation.

The stresses introduced into the pipes by this treatment can be imitated by repeated-folding tests, and these tests taken on tubes of $\frac{1}{8}$ in. bore and 0·15 in. wall thickness showed an irregular behaviour of the special Cr-Mo-Si steels. Some test pieces could be bent back and forth like normal boiler plate, whilst others cracked, even after a moderate primary deformation, as soon as it was attempted to bend them back. The difference was very pronounced, and no intermediate stages could be observed. The composition and mechanical properties of the material provided no clues to the causes of this difference. A greater severity of primary deformation before straightening increased the crack sensitivity of the specimens, but did not resolve the differences in behaviour. It could be shown, however, that the grain structure of the material has a decisive influence. All the specimens which broke easily showed a decarburised, mostly coarse-grained, surface layer of considerable thickness (0·01 to 0·03 in.), whereas in sound test pieces, decarburised surface layers were hardly noticeable or entirely absent. Sensitive materials showed a typical crystalline structure, as is usual in plain high-carbon steels, and sound materials a needle-shaped crystallite structure or a crystalline structure without any decarburisation.

Stress-annealing after bending does not guarantee any improvement of the tube, since the crack may already have been initiated due to the spring back of the pipe. Suitable careful heat treatment of the rolled tube, however, improves the material and may prevent cracking. Undue compression should be avoided during mechanical working so that the material shall remain resistant to subsequent tensile stresses. Finally, it is important that the profile of the bending rolls should be suitably chosen so that deformation of the pipe cross-section may be minimised.

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PHYSICS

The Nature of the Electric Spark

By J. M. MEEK. (From *Bulletin of The Liverpool Engineering Society, Great Britain*, Vol. 23, No. 7, 1950, pp. 11-35, 7 illustrations.)

A SPARK can be defined as an unstable, irreversible and transient phenomenon which marks the transition from one more or less stable condition of current flow between two electrodes in a gas to another more or less stable condition. Knowledge of the nature of the spark discharges has advanced considerably during recent years and the present paper gives a summary of modern views concerning the spark mechanism and of recent experimental research.

Recently Raether[1] and the author[2,3] have independently put forward new theories for the formation of sparks in gaps of a length of 1 cm or more in gases at atmospheric or higher pressure. These theories are essentially similar and are now frequently referred to as " the streamer theory " of the spark. In this theory, the space-charged field produced by individual avalanches is considered, and it is proposed that when this field becomes comparable with the externally applied field, a transition from the avalanche to a streamer occurs. This streamer, known alternatively as a " leader stroke," then grows across the gap to form a conducting filament, namely the spark channel, joining the two electrodes. The external circuit then discharges through the spark channel. The streamer theory depends on ionisation processes in the gas, including photo-ionisation and ionisation by electrons, and is independent of secondary cathode processes, in agreement with experimental observation.

In a spark channel in atmospheric air, the gas density is initially the same as that of the surrounding gas, whereas the gas temperature and pressure in the channel may be forty times higher than the surrounding atmospheric values. In an arc channel in atmospheric air, the gas pressure is the same as that of the surrounding gas, but the gas density may be about one-twentieth atmospheric and the gas temperature about twenty times the atmospheric value. Although in the arc channel metal vapour is frequently present in appreciable quantities, the spark channel initially consists of an ionised column of gas only, and a time of several microseconds may elapse before metal vapour from the electrodes diffuses into the spark channel.

Studies of the appearance of spark channels as a function of time have been made at Liverpool by Holtham and Prime[4], using an electro-optical shutter embodying a Kerr cell unit. Other types of electro-optical shutters have been and are still being investigated at Liverpool by Prime, Turnock and Brown. From a knowledge of the current density and the voltage gradient in the spark, the instantaneous power input can be calculated. For a spark carrying 500 amps, the power input at a time of 1 microsecond from breakdown amounts to about 1,000 kw/cm^2. Such information is clearly of interest in relation to the influence of sparks in the ignition of gases, and future work on this subject is being planned.

The measurements on spark channels are being extended at Liverpool by Allen, to sparks carrying peak currents ranging up to more than a half-million amps, several times higher than the highest recorded lightning current, and an impulse generator with an output discharge capacitance of 70 microfarads at 25 kV has been built for this purpose by Durnford and Reynolds[5].

(1) H. RAETHER, *Zeitschrift für Physik*, Vol. 107, 1937, p. 91 ; Vol. 110, 1938, p. 611 ; Vol. 112, 1939, p. 464 ; Vol. 117, 1941, p. 375, p. 524.
(2) J. M. MEEK, *Physical Review*, Vol. 57, 1940, p. 722.
(3) L. B. LOEB and J. M. MEEK, *The Mechanism of the Electric Spark*, Stanford University Press, 1941.
(4) A. E. J. HOLTHAM and H. A. PRIME. Accepted for publication by the Proceedings of the Physical Society.
(5) J. DURNFORD and P. REYNOLDS. In course of preparation.

POWER PLANT ENGINEERING

Wind Power Plants for Telephone Exchanges

By M. OBERHOLZER. (From *Hasler Mitteilungen, Switzerland*, Vol. 8, No. 3, December, 1949, pp. 83-84, 2 illustrations.)

FOR some time, a small wind power generator of 300 W capacity, designed to supply telephone exchanges in remote villages where the ordinary electricity supply is not permanently available, has been in successful operation.

This generator is rotatably mounted on a tubular mast on the roof of the building housing the telephone exchange. The wheel has two variable pitch blades and is mounted directly on the shaft of the generator, which runs in two strong ball bearings. The pitch is governed by the speed of the generator.

A minimum current and reverse current relay connect the generator to a battery when the generated voltage is sufficiently high ; a voltage regulator controls the shunt excitation to ensure that at high speeds the generator is not overloaded. The generator already begins to charge the battery at a wind velocity of 3 m/sec, and its output at full speed (approximately 10 m/sec wind velocity) is 6 A at 55 V. If, during prolonged periods of wind, the battery voltage reaches 55 V, an artificial load is automatically connected up, which prevents the battery from being overcharged. When the voltage again decreases to 52 V, this artificial load is automatically disconnected.

During one year's operation, this generator has produced a monthly average of 250 Ah at 50 V, i.e., 12·5 kWh.

METHOD FOR MEASURING THE MASS FLOW OF GASES OR LIQUIDS

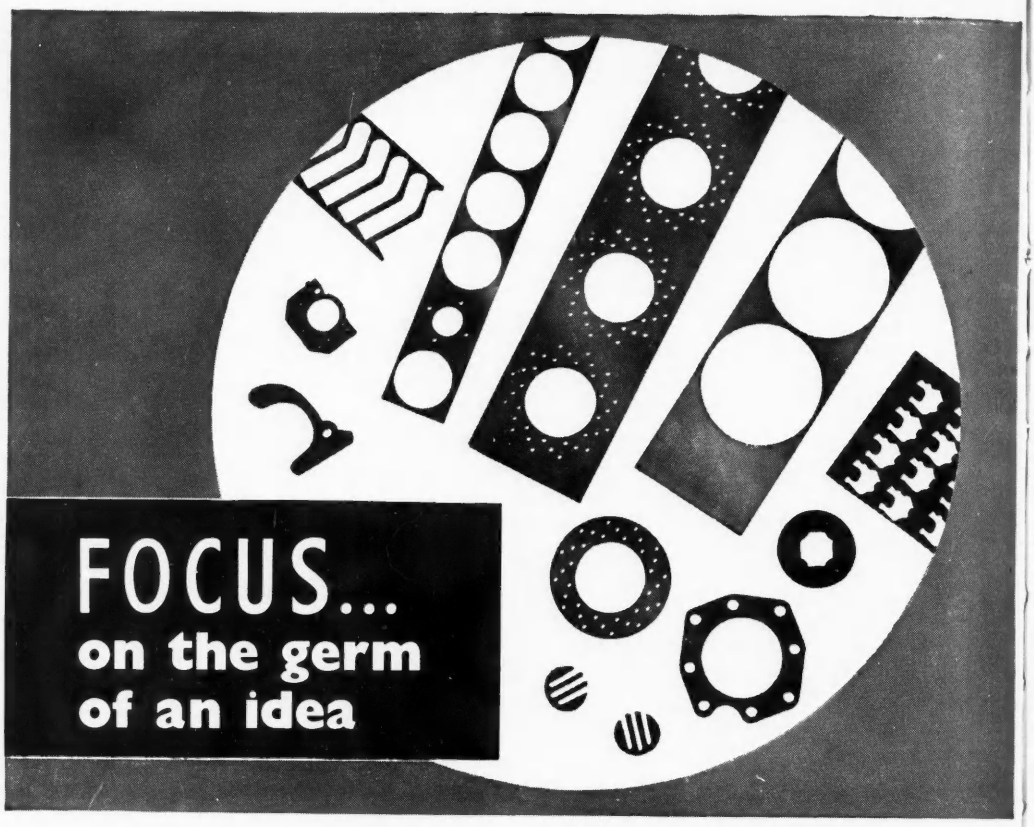
(*Concluded from page 180*)

The short transition through the secondary region is not noticeable in liquid measurements, unless in connection with the rotor peripheral velocity the main flow intensity becomes exceptionally high. The third measurement range can be observed in gas measurements, when mass flow meters specially developed for this purpose are used.

CONCLUSIONS

In the course of the tests of these flow meters it has been found that (1) particularly careful design of the inlet and outlet passages connecting with the main flow is necessary, in order to limit pressure losses ; (2) as a result of the use of sealing surfaces perpendicular to the rotor axis, the rotor of type No. 2 absorbs comparatively more energy than that of type No. 1 with cylindrical sealing surfaces, so that the latter is actually preferable.

The results of the tests with the two prototypes described above are proof of the practical reliability of these flow meters. In addition to being able to measure steady flow with approximately the same accuracy as the usual contraction-type flow meters, the new designs have the advantage of giving measured results which are independent of density variations or pulsations occurring in the flow. Against these advantages is the drawback of the somewhat more complicated construction and the use of an electric motor drive. The mass flow meter should be able to yield useful results in cases where particularly reliable measurements are of importance and where these cannot be achieved with the standard types of equipment.



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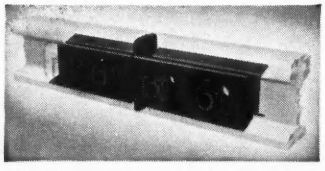

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NEWS OF THE MONTH

PERSONAL

Mr. R. G. Braithwaite, M.I.C.E., has been elected chairman of the Midlands Association of the Institution of Civil Engineers in succession to Mr. S. J. Davies, A.M.I.C.E.

Major G. P. Bulman has been elected president of the Royal Aeronautical Society.

Mr. W. P. S. Cockle, B.Sc.(Eng.)(Lond.), M.I.C.E., has been appointed principal executive assistant to the Transport Executive.

Mr. W. Hadley, general sales manager of E. H. Jones (Machine Tools) Ltd., has been elected to the Board. **Mr. R. G. Longley** has been appointed European technical manager.

Dr. R. Hargreaves has joined the Tiltman Langley Laboratories Ltd., Redhill, Surrey, as chief metallurgist.

Mr. J. Harvey, A.M.I.Mech.E., has been appointed station superintendent at Woolwich generating station of the British Electricity Authority.

Mr. T. S. Kilpatrick has been appointed a director of the Workington Iron and Steel Co., Workington, a subsidiary company of the United Steel Companies Ltd.

Mr. K. G. Lewis has been appointed Principal of the Kenrick Technical College, West Bromwich.

Mr. A. W. C. McArthur, M.I.E.E., M.I.R.I., works manager of the North Woolwich cable factory, has been appointed a director of W. T. Henley's Telegraph Works Co., Ltd.

Mr. R. W. Stuart Mitchell, M.Sc., A.M.I.C.E., A.M.I.Mech.E., has been appointed chief development engineer of the Diesel Engine Division of the English Electric Co., Ltd., with headquarters at Rugby.

Sir Rowland Smith, M.I.Mech.E., F.R.S.A., has been elected chairman of the Ford Motor Co., Ltd., Dagenham, Essex.

Mr. T. Watson, sales director of General Refractories Ltd., Sheffield, has been elected president of the Silica and Moulding Sands Association.

Mr. F. M. G. Wheeler, B.Sc., A.M.I.Mech.E., M.I.Loco.E., has been appointed Head of the Railway Sales Division of British Timken Ltd., Duston, Northampton.

Dr. W. B. Whitney, M.I.C.E., M.I.E.E., has been appointed assistant director of research of the British Electrical and Allied Industries Research Association, Thorncroft Manor, Dorking Road, Leatherhead, Surrey.

Mr. W. F. S. Woodford has been appointed acting secretary of the Institution of Production Engineers, following the resignation of the secretary, Major C. B. Thorne, M.C.

THREE NEW DIRECTORS FOR BOARD OF A.B.O.E.

Three new Directors have been appointed to the Board of Associated British Oil Engines Limited.

They are Dr. J. W. Bondi, an Overseas Director of Associated British Oil Engines (Export) Limited and a Director of British Oil Engines S.A., ABOE's Belgian Sales company; Mr. Bosworth E. Monck, A.M.I.Mech.E., M.Inst.F., Sales Director of Henry Meadows Ltd.; and Mr. J. T. Rymer, M.I.P.E., General Manager and a Director of Mirrlees, Bickerton and Day Ltd., of Stockport, and a Director of Associated British Oil Engines (Export) Ltd., and of Associated British Oil Engines (Marine) Ltd.

The whole Board now comprises: Messrs. Alan P. Good, A. C. Geddes, R. W. Richards, A. P. Quarrell, E. M. Benjamin, J. Calderwood, W. W. Hamill, B. E. Monck, J. T. Rymer and Dr. J. W. Bondi.

JET FLYING-BOAT PROJECT

The first all-jet civil flying-boat has been designed by Saunders-Roe Limited of Cowes, Isle of Wight, and is being considered by Tasman Empire Airways of New Zealand. Known as the *Duchess*, the new flying-boat is designed to use six de Havilland turbojet *Ghost* engines and swept-back wings like those on the *Comet*.

Travel in the jet flying-boat, it is claimed, may cost little more than a penny a passenger-mile. This is the estimated cost for routes of 1,300-1,500 miles, while for a stage distance of 2,000 miles, the designers claim a cost of twopence per passenger mile. The designed all-up weight is 130,000 lb. with a cruising speed of 500 miles an-hour, carrying 74 passengers.

At present Saunders-Roe Ltd. are completing the construction of the three *Princess* giant flying-boats due to make their first flights next year. B.O.A.C. expect to fly them initially on the South African or South American routes, and one of the *Princess* boats may be entered for the London-New Zealand Air Race in 1953.

NEW JETS ON SHOW

Air Force fighters and bombers at the Royal Air Force Display in July will be flying more than twice as fast as those which flew at the last Air Force Display held at Hendon in 1937.

The Hendon Display took place when the jet engine had hardly progressed beyond the design stage. It remained a closely guarded secret for a further five years. This year, Britain's jet-engined fighters, Gloster *Meteors* and de Havilland *Vampires* will be showing off their aerobatic skill at speeds of 500 to 600 miles-an-hour. Also on show will be the R.A.F.'s new fighter, the de Havilland *Venom*, which has a new wing design.

Like the *Venom*, Britain's first jet bomber the *Canberra* will be making its second public debut. They were first shown at the S.B.A.C. Display at Farnborough last year. Although the performance of the *Canberra* is still secret, its manoeuvrability at high speeds indicates that it would be a match for any attacking fighter.

The S.B.A.C. Exhibition and Flying Display is being held this year at Farnborough, Hampshire, 5th-10th September. As usual, this will be the occasion for the debut of the latest prototypes, whose first appearance are being reserved for the occasion.

Czech Cars in Canada. The Czech motor-car industry is making a determined bid for the Canadian market. While only 56 Czech cars were shipped to Canada last year, the arrival of 1,600 is now shortly expected. There are two small Czech cars, the Skoda in the British Austin class which sells at around 1,600 dollars and the air-cooled, rear-engined Tatra car which sells for 2,195 dollars. A Dealers' organization is at present being set up by the Czech motor-car industry.

The Eriksberg Shipyard of Gothenburg, Sweden, will build diesel engines on licence from William Doxford & Co., Ltd., Sunderland.

The largest Floating Crane built in Germany since the war has just been delivered to the Oslo Port Authority. Built by the Luebecker Maschinengesellschaft, Luebeck, the crane has a lifting capacity of 100 tons and a maximum lifting height of 35 metres above water level. The pontoon is 35 metres long and 16 metres wide. Two stern propellers give it a speed of 3 to 4 nautical miles.

Europe's largest Milk Powder Factory has been opened at Beilen, Holland, by the Dutch Minister of Agriculture, Mr. Sicco Mansholt. It is owned by the "Central Milk Products Industry Domo" and has a daily capacity of 126,000 litres of fresh milk. It is equipped with modern installations, and only four workers are needed to make 50,000 litres of milk into powder in eight hours.

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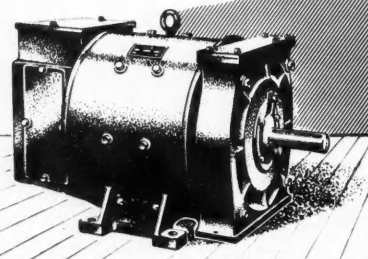

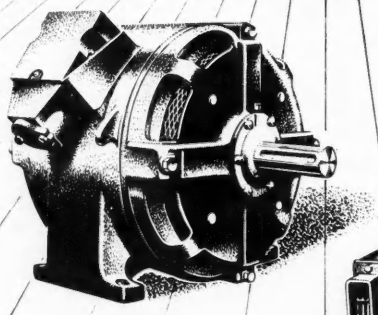

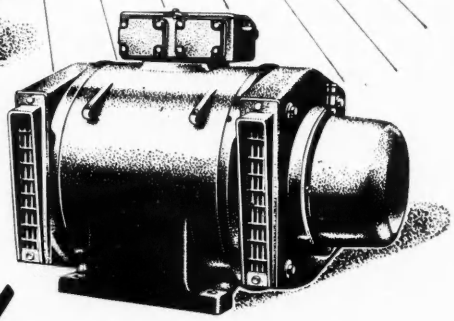

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Industrial Electronics. A series of four lectures in the field of Industrial Electronics will be given by members of the staff of the Mullard Electronic Research Laboratories at The Engineering Centre, 351, Sauchiehall Street, Glasgow, C.2.

These lectures will be on Thursday evenings, commencing 7.30 p.m. After the lecture on May 11th by B. E. Noltingk, Ph.D., A.Inst.P., on Ultrasonics, H. A. Dell, Ph.D., will deliver a lecture on Some Industrial Measuring Instruments on June 8th. Further lectures will be given on September 14th by J. A. Jenkins, M.A., on Photoelectric Devices, and on October 12th by H. A. Dell, Ph.D., and B. E. Noltingk, Ph.D., A.Inst.P., on Vibration Test Methods and Measurements.

Change of Swiss Patent Law. The Swiss Government has presented a draft bill to Parliament introducing several changes into the present Swiss patent law. Official examination of inventors' patents, expansion of the patent right in the textile industry and a general prolongation of the duration of a patent from 15 to 18 years are proposed.

BUSINESS NOTES

Change of Address. The address of John Brooks (Lye) Ltd., Victoria Works, West Bromwich, has been changed to Hall Street, Wednesfield, Staffs.

The Sheffield Twist Drill & Steel Co., Ltd., has removed the London office from Thames House, Queen Street Place, E.C.4, to Terminal House, Lower Belgrave Street, S.W.1. The London office carries large stocks of Dormer products for the use of customers within the London area.

The General Electric Co., Ltd., has acquired the system of picture telegraphy previously made and sold by the Unit Telephoto Co., Ltd., of West Nile Street, Glasgow.

Manufacture and selling of the equipment will in future be carried out by the G.E.C. and production of the apparatus has already started in the Company's Telephone Works at Coventry.

The equipment, which is suitable for operation over line or radio circuits, can be included in portable or fixed installations and is in use by the Press in Great Britain.

Considerable interest has been shown in the equipment abroad and with the backing of the G.E.C. world wide organisation, is expected to become a valuable export.

Rhodesian Alloys Ltd., Gwelo, Southern Rhodesia, sponsored by John Brown & Co., Ltd., has completed arrangements for the construction of a new metallurgical plant.

The Technical Information and Documents Unit of the Board of Trade, which is responsible for the preparation of the B.I.O.S. Reports, has been transferred to Lacon House, Theobalds Road, London, W.C.1.

The Institute of Electronics' Exhibition will be held at the Manchester College of Technology from the 18th to 20th July.

The Brush Group have acquired an interest in the Switchgear and Erection Company (Pty.) Limited, of Germiston, Johannesburg. This company has been re-formed and its title is now the Switchgear and Engineering Corporation of South Africa (SECOR). The present range of switchgear will be extended, and the manufacture of small transformers and motors will shortly be introduced.

Mr. L. H. L. Badham will continue as Managing Director. Mr. Joseph White, Chairman and Managing Director of Brush S.A.) Ltd., and Mr. Sydney A. Lane, an executive director of The Brush Electrical Engineering Co., Ltd., are joining the Board of SECOR.

The mechanical uses of permanent magnets have so far received much less attention in the press than those in the electrical field, so that special interest will attach to the article on this subject, to be found in the current issue of "The Nickel Bulletin." Magnetic chucks, torque transmitters, and welding clamps and lens-grinding magnets are among the items described in this article.

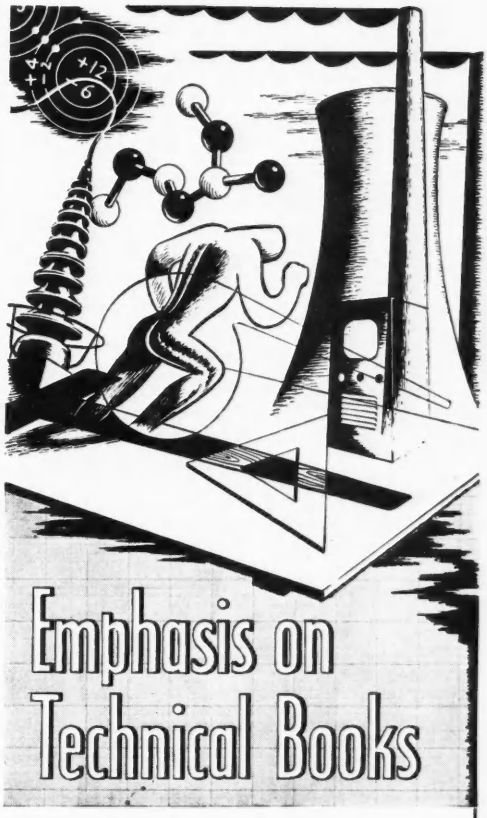
Copies of "The Nickel Bulletin," which also contains a large collection of abstracts of current published information on nickel and its alloys, are obtainable, free of charge, from The Mond Nickel Co., Ltd., Sunderland House, Curzon Street, London, W.1.

HIGH-SPEED REVERSAL OF ELECTRIC MOTORS

A method for reversing a small electric motor in three to four milliseconds has been developed by Jacob Rabinow at the National Bureau of Standards. Designed specifically to meet the need for high-speed reversal of magnetic tapes in the memories of electronic digital computing machines, the technique may prove useful in many other applications.

While it is relatively easy to stop a motor quickly by the use of brakes, rapid starts in either the same or the reverse direction are limited by the low torques which can be obtained by electromagnetic means alone. In the Bureau's rapid-reversal motor, the kinetic energy of the rotor, instead of being dissipated as heat in a brake during deceleration, is converted into potential energy in a spring, which is then used to accelerate the rotor rapidly in the opposite direction.

A small low-inertia two-phase motor operating at 3200 rpm was used. The reversal spring consists of a steel torsion bar approximately 31 inches long and 3/16-inch in diameter. Only one phase of the motor is connected to the a-c power supply; thus the motor will rotate in the starting direction, either clockwise or counter-clockwise. The motor shaft is rigidly connected to one end of the torsion bar, which is equipped at the other end with two positive unidirectional clutches. One clutch prevents clockwise,



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and the other counterclockwise, rotation. If the motor is rotating in a clockwise direction and the proper clutch is engaged, the adjacent end of the torsion bar is thereby stopped; this brings the rotor to a stop in approximately 20 degrees. The potential energy stored in the torsion bar is then returned to the rotor in the form of a counterclockwise impulse. The motor attains virtually full speed in the new direction within about two milliseconds.

Two methods were used in the Bureau's investigation to check the reversing time of the motor. For the simpler test, a black disk provided with a white index-line was mounted on one end of the motor shaft and photographed against a circular scale, using a high-speed motion-picture camera at approximately 4000 frames per second. When the film is run through a projector at the normal rate of 16 frames per second, the behaviour of the disk is easily observed. When the motor is rapidly reversed, the backward speed is, for a short time, slightly lower than the speed before reversal because the ball-bearing assembly does not reverse its rotation as rapidly as does the motor shaft.

A second means of measuring reversal speed, suggested by Frank Shepard of the Shepard Laboratories, involves locking the far end of the shaft and applying direct current to one winding of the motor, while alternating current is supplied to the other winding. Oscillations are thus induced in the rotor-torsion bar. Varying the frequency of the current makes it possible to determine the frequency of torsional resonance. The time required for the rotor to reverse can be easily computed since it is one-half of one period of a torsional oscillation at resonance. The resonance in the system tested occurred at approximately 125 cycles per second, indicating (in a period of 8 milliseconds) a reversal time of 4 milliseconds.

In the experimental model of the rapid-reversal mechanism built at the Bureau, the clutches are operated manually, but it is expected that in normal use the clutch mechanism will be operated by suitable electromagnetic controls. Although the studies thus far have used a small motor of about 1/75 hp, it is anticipated that motors of all sizes could be reversed rapidly by this technique, the speed of reversal being limited only by the mechanical strength of the various parts. Moreover, by use of suitable circuitry, rapid reversal of motors can be obtained without producing large current surges in the electrical supply lines.

It would be entirely feasible to utilize other types of energy-storing devices, such as coil springs or pneumatic cylinders, instead of the torsion shaft. It would also be possible to use a stationary energy-storing system and to locate the reversing clutches between it and the motor shaft.

NEW HYDRAULIC INDUSTRIAL PUMPS

The range of hydraulic industrial pumps introduced by The Plessey Company, Limited, Ilford, Essex, is made in three size-groups which provide a choice of fourteen sizes arranged to give the widest practicable range of capacities. Adoption of the standardisation principle minimises the total number of parts necessary to complete the range while permitting a large degree of interchangeability. Thus, with the exception of gears and bearings, common parts are used in all pumps of each group. The total number of port adaptor sizes has been reduced to three, these being so arranged that the inlet port of a lower group fits the outlet port of the next highest.

The degree of flexibility in mounting is such, it is claimed, that these pumps may be fitted on many installations now utilising hydraulic power, no structural alteration being necessary. All pumps can, for instance, be assembled for either direction of rotation. Still greater flexibility is ensured by the provision of three sizes of elbow and three of straight port adaptors, permitting five directions of porting on both inlet and outlet ports on each pump throughout the range.

Efficiency is increased and maintained by the use of bearings which are hydraulically loaded axially, so reducing side clearance between gears and bearings to the minimum consistent with the maintenance of a satisfactory oil film.

Metallurgical Congress in Italy. The 4th National Congress of the Associazione Italiana di Metallurgia will be held from September 28th to October 1st, at Florence. Two papers, one on "Pure Metals" and the other on "Metallurgical and Engineering Aspects of Machining" will be read and discussed. The third day will be devoted to discussion of papers on subjects freely chosen.

Crane Built of Metal Tubing. The Italian Ferrotubi Company has designed a new light tower-type travelling crane suitable for carrying building materials from the point of operation to the point of use. It is built almost entirely of metal tubing and it can easily be set up and dismantled in a small space only slightly greater than the area of its base.

Berlin Automobile Exhibition, 1950. A great number of foreign as well as German firms will be represented at the 1950 German Automobile Exhibition, to be held from May 27th to June 4th. The United States will lead amongst the foreign exhibitors, being represented by General Motors, Chrysler, Studebaker, Nash, Kayser-Frazer, Overland and Hudson. Next comes Britain with four exhibitors, including Austin and Morris. France will be represented by three firms, including Renault, Italy with two, including Fiat. Austria, Sweden and Switzerland will also be exhibiting. The Ford Motor Company will show examples of both American and German cars.

E.C.A. Dollars for French Industry. The latest Marshall Plan allocation for the French industry included $4,142,000 for the Simca Motor Car Factory and $1,685,000 for the Citroen Company. The E.C.A. has also authorised $1,930,000 to increase production of electric power at Nantes Chevire (Lower Loire).

French Oil Hopes. Engineers of the Societe Nacionale des Petroles d'Quitaine have drilled an oil well in the foothills of the Pyrenees, near Pau. The well spouted oil at the rate of 200 tons a day. The company is now drilling two more wells in the same area and is planning to increase its drilling operations with additional U.S. equipment acquired through the Marshall Plan. High priority for dollar aid has been given to oil exploration work in France.

U.S. officials of the E.C.A. in Paris point out that France now imports nearly 8,000,000 tons of oil for payments in dollars. Oil from the new gusher is being sent to the Shell Company's refinery near Bordeaux, where formerly dollar oil was processed exclusively.

Catalytic Cracking Plant at Fawley. The preparatory work for the construction of the new Esso refinery at Fawley now having been largely finished, the first major stage in the construction of the refinery proper has recently been started with the laying of the foundations of the fluid catalytic cracking plant which, when completed, will produce a million gallons of motor spirit a day and will be the largest in Britain.

The new refinery at Fawley, of which the "cat cracker" is perhaps the most important part, is being built by the Anglo-American Oil Company, Limited. When it is completed, it will produce over five million tons of petroleum products a year and will be the largest oil refinery in Europe.

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 24, Victoria Street, London, S.W.1)

Steel Butt-Welding Fittings for the Petroleum Industry, B.S. 1640, provides for fittings manufactured by various processes which are suitable for butt-welding into pipe-lines. The following fittings have been covered :—

45° elbows, 90° elbows, return bends, 45° and 90° tangent elbows, equal tees, equal crosses, reducing tees, reducing crosses, concentric reducers, eccentric reducers, caps, lapped joint stub ends, 45° and 95° shaped branch pieces (nipples), saddles and split sleeves.

The standard includes complete dimensional tables together with requirements for manufacture and methods of test. Price 7/6.

Ball and Roller Bearing Plummer Blocks, B.S.1642 ; 1950, lays down the leading dimensions which are essential to ensure satisfactory installation and interchangeability of a series of ball and roller bearing plummer blocks for shafts ranging in diameter from $\frac{1}{4}$ in. to 3 in.

The standard relates to the main dimensions ; centre heights, bolt centres, bolt diameters, and maximum overall widths, lengths and heights. It does not deal with design or with the materials from which the plummer blocks are made. Price 1/-.

Graphical Symbols for Fire Protection Drawings, B.S. 1635 ; 1950, gives a comprehensive series of graphical symbols for use on various classes of plans, diagrams, etc., relating to fire protection, combined in some cases with reference letters. They include methods of indicating relative degree of fire hazard, elements of structures, fire protection appliances and other features. Price 3/-

Valves, Gauges and other Safety Fittings for Land Boilers Installations, B.S. 759 ; 1950). In view of the experience gained with the original standard, the restrictions applied to the use of cast iron, bronze and gunmetal, have been reviewed and easements have been made.

This standard deals with safety valves, high-and-low water alarms, stop valves, feed valves, blow-down fittings, water gauges, pressure gauges, test connections and fusible plugs. It does not provide for fittings for calorifiers nor for certain low pressure and hot water boilers. Materials, temperature and pressure limits and method of construction are specified together with formulae for determining the minimum aggregate area of safety valves and the size of springs for spring-loaded valves. Recommendations are included in a foreword in regard to the installation of safety valves, blow-down mountings, fusible plugs and water gauges. Price 3/-.

The Industrial Radiology Group of the Institute of Physics will hold their Summer Meeting in the New Horticultural Hall, Westminster, in connection with the International Congress of Radiology from 24th to 28th July. Papers will be read at meetings on Tuesday to Friday, 25th-28th July, and an Exhibition of radiography, photographs, and general items of interest to those engaged in industrial radiology will be open during the week. Admission to the Exhibition and meetings will be free and open to all. Further details may be obtained from the Secretary, Institute of Physics, 47 Belgrave Square, London, S.W.1.

Nickel Alloy Spring Materials. A new publication issued by Henry Wiggin & Co., Ltd., Birmingham 16, described in some detail the properties which have led to the wide use of their High Nickel Alloys as springs, operating under unusual and exacting service conditions.

A particularly interesting section indicates the possible applications of the Nimonic Series of Alloys for springs operating in high temperatures.

Other materials covered in this twenty-eight page booklet include Nickel, Monel, K-Monel, Inconel and Ni-Span C.

Copies of this publication may be obtained, free of charge on application to :—Henry Wiggin & Co., Ltd., Wiggin Street, Birmingham 16.

The Properties of Aluminium and its Alloys. This Information Bulletin, contains specified compositions and minimum mechanical properties of aluminium and aluminium alloys together with other data. The present edition constitutes a very thorough revision of the initial publication (published by the Wrought Light Alloys Development Association and dealing only with the wrought materials) and is timely with the publication of most of the new series of British Standard Specifications for aluminium and aluminium alloys in both the cast and the wrought forms. By permission of the B.S.I., quotations from these Specifications (BS. 1470-7 and 1490) are made ; some of the data are subject to amendment where they are quoted from Specifications not yet published.

In addition to the tables of mechanical properties there are notes on characteristics affecting design, a section on durability and a section dealing with working methods in relation to the properties of the materials. It is emphasised that on many of these points—and particularly working methods—the reader should refer to other Bulletins dealing each with specific methods of treatment, manipulation or finishing.

Appendix I gives average weights of aluminium alloys in standard forms. Appendix II relates the new nomenclature to that of older Specifications such as BS. Aircraft, DTD and BS/STA7.

The text concludes with a selected bibliography divided between literature on fatigue properties and high temperature properties, and there are also the customary lists of A.D.A. Information Bulletins and A.D.A. Reprints of Papers and Articles. These, as already indicated, amplify many points in the present highly compressed text.

A new rubber pinch valve for use in the mining, paper, chemical, oil, food and beverage industries, has been developed by the mechanical goods division of United States Rubber Company, Rockefeller Center, New York 20.

The new valve will outwear metal when installed in pipelines carrying abrasive or corrosive mixtures, the company said. Its flexibility will offset misalignment in pipes. No packing or repacking is required. It absorbs vibration, eliminates " water hammer," and affords a positive seal in the closed position. Its metal parts can be refitted to new valve bodies thereby reducing replacement costs, and it will break up galvanic action in metal lines.

It can be used on pipe lines carrying such materials as pulverized coal, mine water, anthracite and bituminous screenings, concrete, wood chips, various kinds of chemicals, paper stock, foods and beverages.

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BOOKS RECEIVED

Antenna Theory and Design. By H. Paul Williams, Ph.D., A.M.I.E.E., Sen.M.I.R.E. Volume One Foundations of Antenna Theory. 142 pp. PRICE 21/-. Volume Two The Electrical Design of Antennae. 522 pp. PRICE 63/-. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2.

Reinehart's Mathematical Tables, Formulas and Curves. Compiled by Professor Harold D. Larsen. 264 pp. PRICE : 15/-. Publishers : Chapman & Hall, Ltd., 37 Essex Street, London, W.C.2.

Engineering Drawing and Drawing Office Practice. By P. S. Houghton, A.M.I.Mech.E. 276 pp. PRICE : 15/-. Publishers : Crosby Lockwood & Son, Ltd., 39 Thurloe Street, London, S.W.7.

Mechanics of Flight. By A. C. Kermode, O.B.E., M.A., F.R.Ae.S. 372 pp. PRICE : 18/-. Publishers : Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2.

Pump Questions and Answers. Covering the Construction, Application, Operation, Installation, Maintenance, and Troubles of Centrifugal, Reciprocating, Regenerative, Rotary, and Vertical Turbine Pumps. By Roy Carter, Igor J. Karassik and Elliott F. Wright. 346 pp. PRICE : 42/6. Publishers : McGraw-Hill Publishing Co., Ltd., Aldwych House, London, W.C.2.

The Mechanics of Engineering Soils. By P. Leonard Capper and W. Fisher Cassie. With a Foreword by L. Scott White. 270 pp. PRICE : 21/-. Publishers : E. & F. N. Spon Ltd., 15 Bedford Street, London, W.C.2.

Definitions and Formulae for Students. Applied Mechanics. By E. H. Lewitt, Ph.D., B.Sc., A.M.I.Mech.E. 44 pp. PRICE : 1/-. Publishers : Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2.

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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

● *NEW METALS*

During the last few years, remarkable progress has been made in the development of processes for the production on a greatly increased scale of hitherto less familiar metals, such as titanium, tantalum, and zirconium. Titanium combines high strength with relatively low weight and good corrosion resistance. A survey of the properties of titanium alloys has recently been made in the United States, but much additional research will be required before the possibilities of this metal can be fully assessed. Potential supplies of titanium are abundant. Tantalum will be remembered for its use as a material for electric lamp filaments in the early days of the electric lamp industry. Recently, tantalum was found to provide a useful substitute for columbium when, as a result of the rapidly increasing use of the latter as stabilising constituent in stainless steels, a shortage of columbium occurred. Zirconium developments are still in the initial stage. The metal is one-fifth lighter than plain carbon steel, but four-tenths heavier than titanium. High corrosion resistance is claimed to be an outstanding characteristic common to the aforementioned metals, and the result of comparative corrosion tests, just announced, are therefore of more than ordinary interest. Acid corrosion resistance of tantalum was found to be complete with regard to the reagents investigated; columbium showed high resistance, except to hot concentrated hydrochloric acid and sulphuric acid, while zirconium and titanium were found to be comparatively least resistant.

● *NEW SPARK PLUG DESIGN*

A novel type of spark plug is being manufactured in America, which is claimed to reduce petrol consumption and to be less liable to fouling. This spark plug fires the charge in the cylinder by a multiple flame jet, the bottom of the plug forming a baffle around the centre electrode. During the compression period, gas mixture penetrates through five holes in the baffle. When the plug fires, this gas explodes first, sending jets of flame out through the holes. It is claimed that it is this "pilot" explosion which keeps the plug clean, because any oil that has penetrated inside the baffle is blown out each time the plug fires. The makers state that with this plug, engine mileage per gallon is increased some ten per cent, owing to the improved combustion obtained. Fouling is claimed to be reduced to such an extent that users report up to 25,000 miles between spark plug overhauls. Outer casing, micrometer adjustment nut, and centre electrode sleeve are made of Monel metal, while the centre electrode itself is made of D-nickel.

● *SINGING TUBE REPLACES FAN*

The fact that sound waves can be utilised to produce mechanical forces has long been known. However, its exploitation has hitherto been largely confined to the field of acoustics. An interesting example of the utilisation of the mechanical effects of sound waves in an entirely different field is provided by the "singing tube"—heretofore regarded merely as a scientific curiosity. Briefly, the singing tube is a relatively long tube with a closed bulbous enlargement at one end, into which is telescoped another tube of smaller diameter, the end facing the bulb being open and the other end being closed. By heating the bulbous portion—for instance, by the external application of a flame—the apparatus is made to vibrate and to emit a musical tone. Hence the name "singing tube." A recent American patent describes how the vibrations of a battery of singing tubes can be used to induce the flow of air and thus to act as a fan. It goes without saying that such a fan effect is limited to very small capacities. However, as shown in the patent specification, it is large enough to operate sealed combustion-type heaters for space heating in airplanes. For obvious reasons, a device for muffling the singing sound will have to be incorporated.

● *ELECTROSTATIC GENERATOR*

Two electrostatic generators developing 20,000 volts and 6,000 volts respectively have been developed for the purposes of the U.S. Army. They are required in the application of portable equipment using electron picture tubes. The generators are driven by clockwork and operate for 19 minutes on a single winding. The generator rotor is made up of plastic laminated with metal foil and revolves between charged insulated metal plates. The voltage is built up by electrostatic influence on the metal foil, and is picked off by

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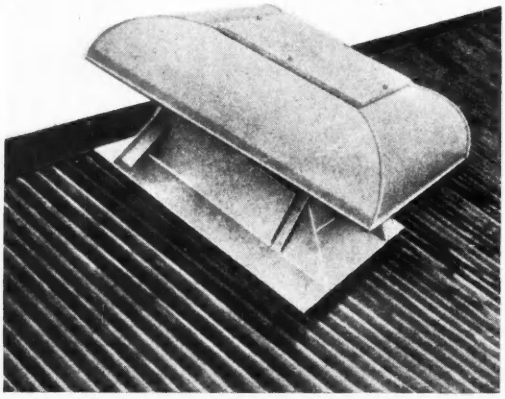

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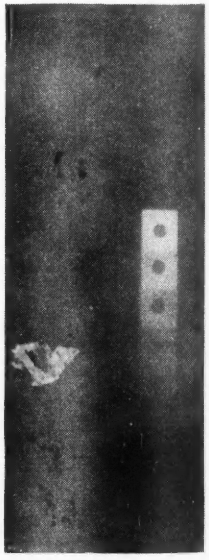

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brushes and conducted to the insulated metal plates. After these plates have become initially weakly charged by an auxiliary friction-type electrostatic generator, the process of building up a voltage and passing it to the plates is continuous as long as the rotor is in motion. The charges accumulating on the plates constitute the source of the high voltage. The generator weighs only about 10 lb.

In a somewhat different design recently developed in France, the machine is housed in a pressure-tight casing filled with a compressed gas which serves as dielectric medium. It is stated that machines of this type can be used either as generators or motors, and the astonishing —but apparently quite reasonable—claim is made that machines of this type can develop powers exceeding 2,000 kW per cubic metre of space occupied. It may be added that this is not, of course, the first time that the use of electrostatic motors has been suggested.

● *HOT-ROLLED CAST IRON*

Hot-rolled cast iron made in a rolling mill from cast iron billets is claimed to possess many attractive physical properties (see *The Engineers' Digest*, October, 1949, pp. 348-350), but, as far as is known, such a rolling process has never passed the experimental stage. In a recent report published by the British Cast Iron Research Association on this subject, the author refers to a recent visit to Germany, where some lengths of rolled strip and bar were examined by him. The report contains the results of the examination of samples brought back to this country. The fact that the British tests indicated ultimate tensile strength values as high as 65 to 72 tsi would serve to confirm the claims made in this respect by the German metallurgists responsible for the production of this material. In view of the increasing application of clad materials, particularly in connection with chemical process plant, it is significant to note claims made to the effect that by rolling cast iron in conjunction with steel, a firmly bonded composite material can be obtained.

● *GAS-FILLED ELECTRIC POWER TRANSFORMER*

Because of its inflammability, oil as insulating medium in electrical power equipment has never been considered as anything more than a necessary evil, and particularly in switch gear design there have been some far reaching attempts in recent years to restrict, if not to eliminate, the use of oil. The latest development in this direction is the evolution of hermetically sealed dry-type power transformers, in which core and coil elements are surrounded by an inert gas which serves both as dielectric and as a means of dissipating the heat losses released internally. This development would have been practically impossible but for the advent of new types of insulating media capable of withstanding higher temperatures than were hitherto found admissible. The substances used for this purpose are silicone resins, which are inorganic compounds in which carbon is replaced by silicon, resulting in a compound of greatly increased heat resistance.

By using silicone resins in combination with such solid insulators as porcelain, mica, asbestos, etc., temperatures as high as 180° C are permissible in the internal parts of the transformer coil. Owing to the employment of nitrogen as the inert gas, oxidation of the silicone compounds is effectively prevented. Studies are already being made to effect further improvements by employing either freon or sulphur hexafluoride as inert medium. The inherent possibilities of the latter substance may be gauged from the fact that at pressures of two to four atmospheres, its electrical insulating strength is comparable to that of transformer oil.

● *ACOUSTIC PROPERTIES OF STRUCTURES*

A study of the acoustic properties of models of concert halls and similar structures calls for the use of sound having wave lengths in proportion to the size of the scale model. This requirement can be met by employing high-frequency sound waves, including ultrasonic waves. It is interesting to note that this method of determining acoustic properties has been employed for the first time by research workers of the Armour Research Foundation of the Illinois Institute of Technology, Chicago. In this test, the acoustic properties of a scale model 1/20th the size of a proposed music pavilion were explored by employing sounds of twenty times the frequency of an average orchestra's music, and probing the acoustics of the structure by means of a microphone sensitive to high-frequency sound. This investigation was extended to a study of " reverberative times " by measuring the rate of sound decay from various parts of the audience area at various frequencies. The sound absorption produced by the presence of an audience could be simulated by strewing very fine glass wool over the seating area of the model.

● *OIL MIST LUBRICATION*

In the oil mist lubrication system, the lubricating oil is atomised by means of compressed air and the air-oil mixture or " oil mist " is then piped to the bearing to be lubricated. This system, which is still in the field-test stage and not yet commercially available, is mainly intended for use on machine tools. It is stated that operating data have been collected since 1938 on a considerable number of installations and that remarkable results have been obtained on high-precision machine tools, particularly grinders with

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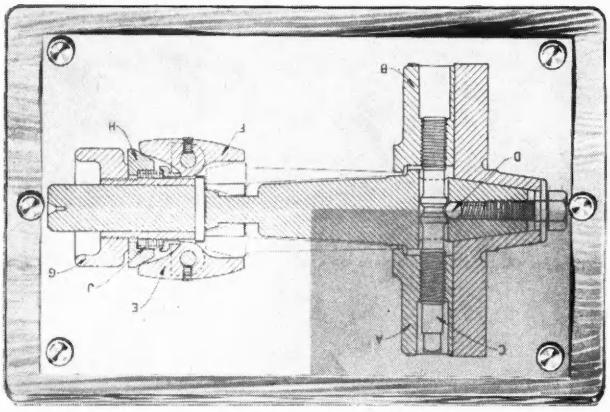

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spindle speeds of well over 100,000 rpm. With regard to the application of oil mist lubrication to heavy duty grinder spindles, the originators of the system point out that due to the feature that the oil mist is supplied to the bearings at pressures in excess of atmospheric pressure, dust or abrasive material cannot enter the bearings. Owing to this feature, bearing life in some instances could be extended to as much as 7,000 hours, as compared to some 300-400 hours prior to application of the new system. An essential prerequisite to successful performance must be the production of an oil mist composed of microscopic particles which stay in suspension and can be conveyed through pipe lines without settling out prematurely. The lubricator producing the oil mist appears to be a compact unit equipped with the requisite air pressure control gear, to be located at the base of the machine to be served. Presumably, protective means are, or will be, incorporated to exclude the possibility of oil mist explosions, should such a danger be real. It is suggested that the use of an inert gas might be considered in certain circumstances.

● *STUDY OF FOAMS AND BUBBLES*

The efficient production of foams is a matter of considerable importance in many industrial processes, as is the effective suppression of foaming in others. In the absence of an adequate fundamental theory, these problems hitherto had to be approached on an empirical basis. Under the sponsorship of the Office of Naval Research of the U.S. Navy, the Stanford Research Institute undertook to conduct fundamental studies which would place the prediction of the foaming characteristics of surface-active agents on a firmer theoretical basis. Through the application of some new techniques, a consistent picture of foaming behaviour could be obtained, and one entirely new relationship between surface tension and foaming was discovered. The effects of impurities upon the foaming properties of surface-active agents were also explored. Continuation of this work should provide useful knowledge applicable to a variety of fields, including those of emulsification, flotation, secondary oil recovery, and perhaps also that of boiler water conditioning.

● *HOT SPRAYING OF BITUMINOUS PAINT*

Bituminous paint is used extensively in ships for interior spaces difficult of access, such as double-bottom ballast tanks, etc. The preferred mode of application of the bituminous paint is in the hot state, and hot-spraying equipment has been used in which the hot bituminous paint is supplied to the spray pistol through a flexible hose. A novel system, the development of which has just been announced, employs a spray pistol into which the bituminous material is fed in the form of a wire of 3 mm diameter, resembling a metal wire-type spray pistol. Immediately prior to reaching the spray nozzle, the bituminous material is liquefied by electric heating, and is subsequently ejected through the nozzle by compressed air, likewise electrically heated. This " wire " spray process is claimed to be of especial value on repair jobs ; moreover, when working in constricted spaces, it will allow even the most awkward spots to be reached. A further claim made for this method is that bubble-free paint coatings as thin as 1 mm—that is, about one-half the usual thickness—can be produced, thus affording a considerable saving in paint.

● *FLEXIBLE VARNISH COATING*

A varnish which remains flexible after baking should prove a useful impregnating medium to apply to transformer coils and other similar coils, as the leads can be bent and twisted without cracking the varnish after it has been cured. The new varnish, for which this feature is claimed, is said to possess also excellent bonding characteristics, high dielectric strength and exceptionally high resistance to attack by moisture, transformer oils, mineral oils and solvents. Oil resistance tests conducted at 100° C are reported to have yielded no evidence of varnish deterioration ; nor was there any discoloration of the oil. Although the varnish retains sufficient flexibility, it is stated that it penetrates deeply into coil windings and bakes throughout to form a unified, solidly bonded mass. Baking times vary from 2 to 8 hours according to baking temperature, which ranges from 220° F to 275° F.

● *CONTINUOUS DETERMINATION OF IMPURITIES IN FLUIDS*

Continuous determination and registration of impurities contained in gases or liquids can be effected by measuring their dielectric constant in a special type of microwave refractometer devised at the National Bureau of Standards, Washington. The extremely high sensitivity of the instrument permits operation with small test samples, and it should be readily adaptable to manufacture as a field model, since the component parts and electrical circuits are straightforward and compact. It should be directly applicable as a supervisory instrument in industrial processes. The operating principle is the comparison of two cavity resonators. A test sample is introduced into one of two otherwise identical cavities. The resultant difference in resonance frequency between the two cavities is then a measure of the dielectric constant of the sample tested. In measuring the dielectric constants of gases, and thus to determine gas purity, it is often convenient to use a flow technique, in which a continuous stream of gas is drawn through the test cavity.

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FOR INDUSTRIAL EQUIPMENT

● *CATALYTIC FUME INCINERATOR*

In the production of enamelled wire, the solvents driven from the coated surfaces during the baking process and exhausted from the oven to the outside of the building frequently cause a local air pollution problem. Frequent cleaning of exhaust manifolds and stack is necessary due to partial re-condensation of the solvents after being discharged from the oven. A recently developed catalytic fume incinerator, capable of completely consuming the fumes in exhaust gases from wire enamelling ovens, simultaneously solves air pollution and recondensate problems. The heart of the catalytic fume incinerator is the catalyst " element," which has the appearance of a metallic air-filter mat. When gases containing a mixture of vaporised hydrocarbon and air at a temperature of approximately 500° F or higher come into intimate contact with the catalyst, the hydrocarbons and the oxygen of the air combine to form the usual products of complete combustion, namely, water vapour and carbon dioxide, with heat liberation taking place. There is no addition to or detraction from the catalyst. The presence of the latter merely reduces the self-ignition temperature of the fumes to approximately 500° F. There is normally no visible evidence of combustion. Catalytic incinerators would appear applicable to many other processes of similar nature.

● *PLASTIC-LINED COLLAPSIBLE TUBE*

In view of the important role played by collapsible tubes in modern packaging technique, recent claims made for a new machine for manufacturing collapsible tubes by a radically new process cannot be lightly dismissed. This new machine, it is reported, produces tubes from foil, laminated or coated on both sides with plastic; the layer serving as liner for the finished tube is chosen in accordance with the requirements arising from the properties of the substance to be packaged. Testing of tubes with various liners for different purposes is said to be in progress. Where necessary, the layers of laminated foil can be double or triple wound in the course of tube manufacture to give the tube any reasonable required strength. Also, tubes of this type can be made in virtually any diameter and length desired. Combination of the various foils and films to be used is said to be limited only by the requirement that the plastic films must be heat-sealable for the tube-making operation.

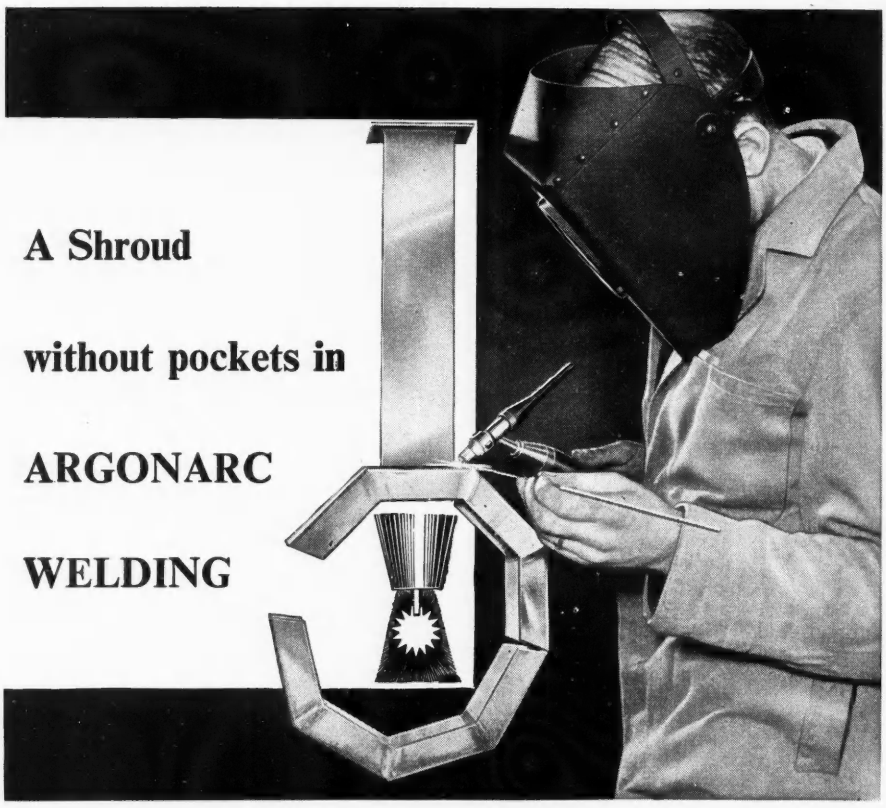
● *LENGTHENING VALVE LIFE*

The possibility of lengthening the life of poppet valves in internal combustion engines by rotation of the valves merits serious consideration even more today than ever before. Imparting rotation to such valves offers at least three important benefits. Firstly, the stem will remain free of deposits, and sticking, which is a common cause of valve burning, will be prevented. Secondly, rotation imparts a light wiping action between the valve face and the seat, making for cooler operating valve heads. The prevention of such face deposits also precludes the flaking-off of some portions, which would allow blow-by and subsequent burning. The third advantage offered by valve rotation is that it prevents prolonged exposure of any one sector of the valve face to a local hot spot on the seat. This produces lower and more uniform valve-face operating temperatures. A reduction in peak valve temperatures will substantially improve valve life. Among the secondary benefits of valve rotation is the prevention of stem scuffing. The rotary motion distributes whatever lubricant there is around the valve stem and prevents metal-to-metal contact which usually is responsible for scuffing. Rotating the valve also cuts down undue tip wear and eliminates the grooving often found on non-rotating valves.

● *FATIGUE PROPERTIES OF NODULAR CAST IRONS*

A recent report by the British Cast Iron Research Association draws attention to important differences in the fatigue properties of ordinary flake graphite cast irons and the new high-strength materials known as " nodular cast irons." Ordinary cast iron is known to have good fatigue properties and a relatively low notch sensitivity. In the case of the nodular material, the limiting fatigue stresses are higher than those of the flake graphite irons, but there is a marked difference in the influence of notches and grooves on the fatigue properties of flake and nodular irons. Tests have shown that stress raisers have a much greater affect on the fatigue properties of nodular irons; and in the case of two nodular irons it was actually found that they were weaker in notch fatigue than either of the pearlitic flake graphite irons tested, although the static strength of the nodular irons was much higher. An extreme case is quoted, in which the notched fatigue strength of a nodular iron of 40 tons tensile strength was only slightly superior to that of a flake graphite iron of 18 tons tensile strength, thus leaving little incentive for the employment of the nodular material under such conditions. However, in static tensile tests, flake graphite irons were slightly weakened by the introduction of a notch or groove, while the nodular material was actually strengthened.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.



A Shroud without pockets in ARGONARC WELDING

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